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OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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CONTENTS

SOCIETY AFFAIRS

Increase of Membership (3); Committees on Power Tests and Standard Specifications (3); Committee on Flanges and Flanged Fittings (4); Society History (4); New York Meeting for October (5); Interesting Meeting to be held in Worcester (5); New Haven Meeting (6); Organization for Meetings at Atlanta, Ga. (6); Proposed Amendments to the Constitution (8); Gift to Prof. Matschoss (8); Applications for Membership (9); Third International Congress of Refrigeration (10); Meeting on Iron and Steel (11).

PAPERS

Pitot Tubes for Gas Measurement, W. C. Rowse 1319

Tests of Vacuum Cleaning Systems, J. R. McColl 1381

Standard Involute Gearing. Majority Report of Committee on Standards for Involute Gears 1405

Discussion: Luther D. Burlingame, Henry Hess, The Committee. 1411

Economics for Central Station Heating, Byron T. Gifford 1421

FOREIGN REVIEW 1427

NECROLOGY 1448

ACCESSIONS TO THE LIBRARY 1449

EMPLOYMENT BULLETIN 1453

OFFICERS AND COMMITTEES 1457

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COMING MEETINGS OF THE SOCIETY

October 10 (Probable Date), *Worcester, Mass.*, Worcester Polytechnic Institute. For details, see page 5.

October 14, *New York City*, 29 West 39th Street. Paper: Stability in Aeronautics, A. A. Merrill. For further details, see page 5.

November 19, *New Haven*, Mason Laboratory, Sheffield Scientific School. Subjects: Industrial Coöperative Research, and miscellaneous papers.

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During the summer months when no meetings are held and committee members are scattered, signs of activity at the Society's headquarters are naturally less marked than at the other and "busy" seasons. During the past summer, however, this has been true only to the extent that there have been fewer wheels of the Society's mechanism in motion. Such as have been kept in operation by committees able to undertake work in the vacation season have run smoothly and well and much has been accomplished.

Of the greatest importance to the welfare of the Society itself is the record of the Increase of Membership Committee, which last June succeeded in raising the membership to more than five thousand. There were 523 members added as a result of the ballot at the Spring Meeting, and since that time 387 applications have been received. The Society is increasing at the rate of one thousand members a year and if this should continue, as it is believed that it will, the coming season will end with a membership of six thousand.

A great deal of work has been accomplished by the committees on Power Tests and on Standard Specifications for the Construction of Steam Boilers, both of which are at work upon voluminous reports. The former committee is revising its preliminary report, as submitted at the last Annual Meeting, and the latter now has the copy for its forthcoming report in completed form in the hands of all its members, for their final

consideration before submitting it to the membership at large. In many respects these two reports are the most important ever issued by the Society. They ought to do much to standardize practice, and improve the efficiency and enhance the safety of power plant apparatus.

The Committee on Flanges and Flanged Fittings has been in conference with manufacturers of valves and fittings with a view to finding some ground on which manufacturers who have large commercial interests and a great deal of money tied up in patterns and castings can come together with a committee desiring to attain a standard to insure the greatest possible safety through a consistent proportioning of the various sizes of flanges and fittings. The situation in this regard was fully outlined in The Journal for August 1912. The changes from previous practice recommended by the committee referred mainly to heavy work and to sizes above 9 inches, where the main considerations are those of safety, and no expense should be considered too great to secure immunity from accident. Their recommendations were such as to secure a reasonable factor of safety throughout the whole range of sizes and a uniformity of dimensions providing for interchangeability of fittings.

Manufacturers at that time were beginning to supply material according to the 1912 U. S. Standard of Flanges and Flanged Fittings recommended jointly by the committee of this Society and that of the National Association of Master Steam and Hot Water Fitters, but a number of them announced that an extra charge would be made where a compliance with the standard necessitated a material change in the existing patterns.

As a result of the conferences our committee has been holding, a tentative schedule has been agreed upon which only awaits formal acceptance by the manufacturers and by the Society.

The new standard fully covers all considerations of safety, but it has not been found expedient to maintain the feature of interchangeability.

Another important undertaking which has been pushed rapidly is the preparation of the Society History, which it is expected will be completed during the present year. Dr. Hutton, secretary of the committee on Society History, is giving this his personal attention and devoting much time to its preparation.

Plans have further been in process by various committees

in charge of local meetings. Definite arrangements are already made for meetings in New York, New Haven, Conn., and Worcester, Mass., the latter being a development resulting from the very successful meetings that have been held in Boston.

NEW YORK MEETING FOR OCTOBER

In New York the opening meeting is set for the evening of Tuesday, October 14, in the Engineering Societies' Building, when Albert A. Merrill of Boston will present a paper on Stability in Aeronautics. Mr. Merrill is the founder of the Boston Aeronautical Society and a lecturer at the Massachusetts Institute of Technology upon the subject of aeronautics.

The question of stability is one of the most important to be studied in aviation. As at present designed the flying machine has no inherent stability, and Mr. Merrill has analyzed the forces involved in order to determine if by a change in design it is possible to increase the safety of the flying machine. He hopes to arouse among engineers a greater interest in aeronautics so that through their efforts it may be put upon a more scientific foundation in this country. Although aviation had its start in America through the work of Langley, Chanute and the Wright Brothers, we are now far behind France in so far as real scientific progress is concerned. The advance there is due to the excellent work accomplished in aeronautical laboratories, notably the Eiffel Laboratory, and there is urgent need of similar opportunities in this country for the study of the science of aeronautics.

INTERESTING MEETING TO BE HELD IN WORCESTER

A new departure is planned in the way of group meetings this fall by the members in the Boston district. It is expected that meetings will be held in other cities in that section, besides Boston, through the coöperation of the Boston members with those residing in the cities selected for meetings. The first of these will be held in Worcester about October 10. The matter is in charge of a special committee of Worcester members, Prof. W. W. Bird, chairman, and it is expected that a considerable number of the Boston members will take advantage of the facilities provided for their attendance. According to the tentative arrangements, a special train will leave Boston about noon, and the party will be conveyed by automobiles to the Worcester

Polytechnic Institute where luncheon will be served. At the afternoon session Prof. David L. Gallup will present a paper on Experiments with Aeroplane Propellers at the Worcester Polytechnic Institute, and there will be an inspection of the shops and laboratories. Those attending the meeting will then be taken in automobiles to the hydraulic testing plant, where there will be propeller tests, and to the administration building of the Norton Company where a paper on Modern Abrasives, their Manufacture and Use, by Aldus C. Higgins, George Jeppson and Charles H. Norton, will be presented. An inspection of the plant and works will follow, and the visitors will be taken by automobiles in small groups to inspect the various industrial plants in Worcester, comprising the steel works, car works, electric light plant, envelope works, loom works, etc. Dinner will be served at the New Bancroft Hotel and the party will return to Boston by special train, arriving about half past ten.

NEW HAVEN MEETING

The quarterly meeting of the New Haven members is announced for November 19, with afternoon and evening sessions, and will, as usual, be held in the Mason Laboratory of the Sheffield Scientific School. One session is to be devoted to the subject of Coöperative Industrial Research, and the other to general papers on a variety of topics.

ORGANIZATION FOR MEETINGS AT ATLANTA, GA.

Atlanta, Georgia, is the tenth city to organize local meetings of the Society. The movement began last June when an organization was effected of the local members of The American Society of Mechanical Engineers in Atlanta and vicinity, with Mr. Park A. Dallis as secretary. This was part of a general movement to bring about coöperation between the various engineering and architectural associations of the city. Besides the membership of The American Society of Mechanical Engineers there are branches of five national societies, the American Institute of Architects, the American Society of Civil Engineers, the American Institute of Electrical Engineers, the American Chemical Society, and the Engineering Association of the South.

It was felt that it would be highly desirable to bring about an affiliation of these interests such as has been done in several other cities to the benefit of engineers individually and of the profession as a whole.

The first step towards this affiliation was an old-fashioned Georgia barbecue held on July 12, attended by fifty-seven members of these various professional bodies, which was in reality an organization meeting. Allen M. Schoen, president of the local section of the American Institute of Electrical Engineers, called the meeting to order and briefly outlined its object and the means employed to bring this about. He called upon James Nisbet Hazelhurst, chairman of the executive committee of the local section of the American Society of Civil Engineers, who spoke of the movement and its advantages, and made a special plea for a wider participation of the technical man in public affairs.

The meeting was also addressed by L. J. Hill, Jr., president of the Atlanta section of the Engineering Association of the South, on the advantages of affiliation by the different technical organizations; by Hal Hentz, representing the American Institute of Architects, on the needs of professional advice in civic matters, such as parks and buildings; by J. S. Brogden of the Georgia section of the American Chemical Society, who spoke of the part of the chemical engineers in such an affiliation; and by Park A. Dallis, Mem.Am.Soc.M.E., on the needs of such an organization. Similar remarks were made by Capt. R. M. Clayton, chief of construction of the City of Atlanta, A. C. Bruce, a prominent architect, V. H. Kriegshaber of the Chamber of Commerce, F. H. Granger, a member of the Franklin Institute and also of the American Institute of Mining Engineers, B. M. Hall, a member of the American Society of Civil Engineers and of the American Institute of Mining Engineers, Prof. G. N. Mitcham of Auburn Polytechnic and also of the Alabama state highway commission, Prof. H. P. Wood of the Georgia School of Technology, Paul Norcross, and others.

As a result of this gathering, an executive committee was chosen to formulate a plan for permanent organization. An executive committee of the Society in Atlanta has also been nominated and will be submitted to the Council at its next meeting for confirmation, thus completing the organization and extending the influence of the Society to this important industrial and

railway center of the South. The local membership is greatly interested in the movement and is disposed to enter into the spirit of the work for the increase of membership which is now going forward so rapidly and which is adding so many prominent names to the roll of members throughout the country at the present time.

PROPOSED AMENDMENTS TO THE CONSTITUTION

The following amendments to the Constitution are to be presented at the Annual Meeting of the Society in New York in December:

C-9 A Member shall be an Engineer or Teacher of Applied Science of thirty-two years of age, or over, and shall have been in the active practice of his profession for at least ten years and in responsible charge of important work for five years, and shall be qualified to design as well as to direct engineering work. Fulfilling the duties of a Professor of Engineering who is in charge of a department in a college or school of accepted standing shall be taken as an equivalent to an equal number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two years of active practice.

C-11 An Associate-Member shall be a professional engineer not less than twenty-seven years of age, who shall have been in the active practice of his profession for at least six years, and who shall have had responsible charge of work as principal or assistant for at least one year. Graduation from a school of engineering of recognized reputation shall be considered as equivalent to two years' active practice.

GIFT TO PROF. MATSCHOSS

A pleasing aftermath of the many satisfactions experienced during the trip through Germany was the presentation of a beautiful repeater watch by the members of the party to Prof. Conrad Matschoss, in recognition of his services during the trip and as an indication of personal appreciation.

When the trip was projected Prof. Matschoss came to America to consult with the officials of the Society about arrangements, and also visited the various cities in Germany to perfect arrangements, and from the time that the steamer touched at Plymouth until the ending of the trip at Munich was with the party constantly as its guide and helper. He was assigned by the Verein deutscher Ingenieure to attend to the many details of management connected with the great undertaking, and the fact that his duties were well performed and the trip made an unqualified suc-

cess afforded him the greatest satisfaction and was, from his standpoint, ample repayment for his devoted services.

As the trip advanced, however, there was an ever-increasing feeling of friendship and appreciation on the part of his American friends which led to a spontaneous movement for some expression of gratitude to Prof. Matschoss, resulting in the raising of a fund and the purchasing of the watch.

The gift is from the famous firm of Vacheron & Constantin at Geneva and was personally selected by Worcester R. Warner, who was in Switzerland immediately after the trip. It is engraved with the words: "Presented to Professor Conrad Matschoss as a token of appreciation and esteem by members of The American Society of Mechanical Engineers visiting Germany in 1913." The watch is a minute repeater, and was sent together with a chain of gold and platinum links, which as Mr. Warner expressed it to Prof. Matschoss, represents the welding of German and American sympathies and interests through his zealous efforts. He voiced also the wish that the watch might repeat its message to Prof. Matschoss with every ringing of its bells. The gift came evidently as a great surprise to Prof. Matschoss, who had written feelingly of his happy remembrances of the wonderful trip.

APPLICATIONS FOR MEMBERSHIP

The Membership Committee have received applications from the following candidates. Any member objecting to the election of any of these candidates should inform the Secretary before October 15, 1913:

ALLISON, DANIEL K., Cincinnati, Ohio
ANDREWS, DAVID, Newton Center, Mass.
BAKER, ORTH K., Tremp, Lerida, Spain
BARON, LEOPOLD, Newark, N. J.
BOSWORTH, WILBUR McK., Pittsburg, Kan.
BOWSER, HOWARD A., Buenos Aires, S. A.
BRICE, ANDREW F., New York, N. Y.
BRYEN, GEORGE J., Duquesne, Pa.
BULLARD, JOHN E., Pt. Washington, N. Y.
BURT, AUSTIN, Waterloo, Iowa
CASE, LYNN B., Hudson Falls, N. Y.
CHILDS, JOHN N., Salt Lake City, Utah
CONROY, RAMON A., Brooklyn, N. Y.
COULTER, JAMES, Bridgeport, Conn.
COWELL, WILLIAM A., Anderson, Ind.
CRANE, ARTHUR M., New York, N. Y.
DALAS, FRANK L., Long Beach, Cal.
DAVIS, GEORGE H., New Orleans, La.
DAVIS, JAMES H., Brooklyn, N. Y.
DIGHE, SHANKERRAO M., Baroda, India
DOWD, BERNARD J., Hartford, Conn.
DU VIVIER, ERNEST H., New York, N. Y.
ESTABROOK, HARRY M., Dayton, Ohio
ELSTER, GURDON G., Montreal, Canada
EVELAND, SAMUEL S., Philadelphia, Pa.
FARR, ARTHUR V., Maplewood, N. J.
FAULKNER, DAVID S., New York, N. Y.
FEISS, RICHARD A., Cleveland, Ohio
GAMMETER, HARRY C., Cleveland, Ohio
GANNON, THOMAS J., Brooklyn, N. Y.
GARDINER, WALTER B., Erie, Pa.
GAY, FREDERICK W., San Francisco, Cal.
GIBSON, CHARLES D., Chicago, Ill.
GREEN, FREDERICK W., Stamps, Ark.
GROVER, CHARLES L., Bethlehem, Pa.
HALL, ARTHUR G., Cincinnati, Ohio

Hanks, Henry M., Narberth, Pa.
Harrold, Frederick F., Munhall, Pa.
Hopkinson, Joseph, Dayton, Ohio
Heinze, John O., Detroit, Mich.
Henn, Oliver L., Cleveland, Ohio
Huestis, Bronson, Brooklyn, N. Y.
Israel, Albert H., New York, N. Y.
Johnson, Allen E., Westmount, Canada
Keller, J. M., Cleveland, Ohio
Kellogg, Morris W., New York, N. Y.
Kennedy, William P., New York, N. Y.
Knight, William, E. Orange, N. J.
Landis, Mark H., Waynesboro, Pa.
Leeson, Charles G., Oroville, Cal.
Lindemann, Walter C., Milwaukee, Wis.
Lord, Caleb W., Philadelphia, Pa.
Lord, Harold S., Athol, Mass.
McClintock, Allan P., Bayonne, N. J.
MacAllister, Robert N., Chicago, Ill.
Mardaga, Louis, Baltimore, Md.
Marx, Henry J., Poughkeepsie, N. Y.
Matschoss, Conrad, Berlin, Germany
Misch, Arthur A., Cleveland, Ohio
Morgana, Charles, Jr., Detroit, Mich.
Nicolai, Charles A., Orange, N. J.
Pfeiffer, Charles G., Philadelphia, Pa.
Pohle, Richard F., Lynn, Mass.
Purchas, Arthur W., Oilfields, Cal.
Richardson, Ammi C., Holyoke, Mass.
Riggs, F. Stanley, Akron, Ohio
Roberts, Richard F., New York, N. Y.
Robison, Charles J., Canton, Ohio
Schroeder, Ernst O., Ann Arbor, Mich.
Sly, Frederick S., Chicago, Ill.
Smith, Donald F., Detroit, Mich.
Smith, Edric B., New York, N. Y.
Stults, William R., Orange, Mass.
Tait, Edward H., Easton, Pa.
Tinsley, George W., Tremp, Lerida, Spain
Tonkin, Ellsworth, Canton, Ohio
von der Werff, J. N., Toronto, Canada
Warner, Raymond C., Pittsburgh, Pa.
Wein, Charles V., So. Bethlehem, Pa.
Wheelhouse, Sidney H., Salem, Ohio
Wilson, Hugh M., Hartsdale, N. Y.

PROMOTION FROM ASSOCIATE

Clarke, Charles W. E., Boston, Mass.
Johnson, Bradley S., Chicago, Ill.

PROMOTION FROM JUNIOR

Coburn, Frederick G., Philadelphia, Pa.
Forté, Harry P., Canajoharie, N. Y.
Gates, Thayer P., Saylesville, R. I.
Posselt, Ejnar, Denver, Colo.
Wegg, David S., Boise, Idaho
Wentworth, R. A., Philadelphia, Pa.

SUMMARY

New applications	80
Promotion from Associate	2
Promotion from Junior	6
	88

THIRD INTERNATIONAL CONGRESS OF REFRIGERATION

At the invitation of the President of the United States the Third International Congress of Refrigeration will be held in Washington and Chicago, September 15-24, inclusive.

The delegates from the foreign countries will be received and entertained in New York City during the week of September 8. An attractive program of entertainment and excursions to points of general and technical interest has been arranged to cover the entire week. During this time the headquarters of the Congress will be in the rooms of the Society.

The delegates will proceed to Washington on Sunday, September 14, and on the following day the formal opening session of the Congress will be held, with a reception by the President and addresses by a number of government officials.

On Tuesday, September 16, those in attendance will proceed to Chicago for a series of meetings to be held in the International Amphitheater, extending over a period of a week. The Congress will be divided into six sections for the consideration of technical and legislative matters of interest to the great industry of refrigeration. The first section of the Congress, devoted to liquid gases and units, is under the direction of William Kent, chairman, and Calvin W. Rice, secretary. Other members of the Society prominently identified with the Congress are: G. T. Voorhees, Assistant Secretary-General, D. S. Jacobus, J. W. Lieb, Jr., Charles E. Lucke, E. F. Miller, and S. W. Stratton.

A number of extended excursions have been arranged to San Francisco, Panama and other distant points in which the delegates will participate subsequent to the Congress.

Programs and full information regarding the Congress may be obtained from J. F. Nickerson, Secretary-General, 431 South Dearborn Street, Chicago, Ill.

MEETING ON IRON AND STEEL

A meeting of the American Institute of Mining Engineers under the auspices of the Iron and Steel Committee of the institute, will be held in New York on October 16 and 17. An informal dinner and smoker will be given on the evening of October 17 to which as well as to the professional sessions the membership of the Society is invited. Papers upon the following subjects will be presented:

Blast Furnace Gas Cleaning, W. A. Forbes

The Quality of Cast Iron as affected by Oxygen, Nitrogen and Other Elements, J. E. Johnson, Jr.

Oxygen in Steel, W. R. Shimer

New Design of Regenerators for Open-Hearth Furnaces, H. F. Miller, Jr.

Shock Tests of Cast Steel, J. H. Hall

Scoria Process, E. Stutz

Briquetting, Felix A. Vogel

Uses and Advantages of Briquettes in Blast-Furnace Practice, Felix A. Vogel

Discussion of the Existing Data as to the Position of Ae_3, H. M. Howe

Determination of the Position of Ae_3 in Carbon-Iron Alloys, H. M. Howe and A. G. Levy

Ae_3, the Equilibrium Temperature for A_1 in Carbon Steel, H. M. Howe

The Divorcing of the Eutectoid in Meteorites, H. M. Howe

Thermal and Microscopical Examination of Professor Howe's Standard Commercial Steels, G. K. Burgess, J. J. Crowe and H. S. Rawdon

Influence of Alloying Elements on the Carburization of Steel, R. R. Abbott
Some Phases of the Practical Treatment of Tool Steel, J. V. Emmons
The Influence of Copper upon the Physical Properties of Steel, G. H. Clevenger and B. Ray
Resistance of Steels to Wear in Relation to their Hardness and Tensile Properties, J. L. Norris
Paper comparing the Records of Crucible Furnaces for Steel Making, J. H. Hall

STABILITY IN FLYING MACHINES

By Albert Adams Merrill

ABSTRACT OF PAPER

Any movement of a flying machine can be resolved into a translation of its mass and a rotation about its center of gravity. The translation of the mass of a flying machine has no relation to stability, which is connected only to rotations about its center of gravity.

These rotations can be referred to the three axes of the machine, vertical, longitudinal and lateral. The vertical rudder controls directly, rotations about the vertical axis, and controls indirectly, rotations about the longitudinal axis, since the former rotations alter the relative tip speeds and hence the relative pressures on the tips.

It is a peculiar characteristic of a curved surface that when the angle between the chord of the surface and the surrounding air stream (called the pressure angle) is increased from 0 deg. the center of pressure moves forward. This tends to increase, still more, the pressure angle, which is dangerous. This dangerous movement is offset by the horizontal rudder. A system known as the converging tandem system, which consists of two surfaces equal in area one behind the other and so placed that the pressure angle of the front surface is always greater than the pressure angle of the rear surface, is longitudinally stable because when the pressure angle increases from 0 deg. the center of pressure moves backward. With such a system the horizontal rudder is not needed as an offset.

In most machines lateral stability is maintained by increasing the positive pressure angle of the tip to be raised. This tends to retard that tip and turn the machine in the wrong direction. This false turning movement is offset by the vertical rudder. It is possible to maintain lateral stability by moving a surface to a negative angle on the tip to be lowered, and this will produce a turning movement in the right direction, hence no offset will be needed.

Present machines are so badly designed that dangerous couples are introduced which have to be offset by other couples introduced by the pilot. That we fly as well as we do is not due to the design of the machine but to the skill of the pilot.

It is possible to design a machine in which the couples introduced are righting couples and in which no offsetting couples are needed. Until such a machine is produced there will be only a small market for the sale of flying machines.

STABILITY IN FLYING MACHINES

By Albert Adams Merrill,[1] Wellesley, Mass.

Non-Member

Flying differs from all other methods of transportation because of the fact that the stability of the machine is dependent upon its speed through the air. In this respect the bicycle is the only other machine used by man which resembles the flying machine. As at present designed, the flying machine has no inherent stability, and this is why flying is dangerous. It is the purpose of this paper to analyze the forces involved in flying in order to determine if, by changing the design of machines, it is possible to make flying safer.

2 For the sake of clearness it is necessary to define certain terms as they are used here:

Span is the distance from tip to tip at right angles to the path

Chord is the distance from front to rear of a supporting surface

Angle of Incidence is the angle between the chord and the horizon

Pressure Angle is the angle between the chord and the path of the machine

Vertical Axis is a vertical line through the center of gravity

Longitudinal Axis is a horizontal line through the center of gravity at right angles to the span

Lateral Axis is a horizontal line through the center of gravity parallel to the span

3 Any motion of a flying machine can be resolved into two components: (*a*) a translation of the whole machine; and (*b*) a rotation of the machine about its center of gravity. Since

[1] Special Lecturer on Aeronautics, Mass. Inst. of Tech., 1912–1913.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

the former has no effect upon stability it will not be considered in this paper. Since there are three axes a rotation about the center of gravity can be resolved into three components. I shall consider, first, rotations about the lateral axis, which are of two kinds: stalling and diving. The former occurs when the bow is raised; the latter, when the bow is depressed.

4 These rotations have a great influence upon safety in flight not only because they throw the machine away from a safe horizontal position but, particularly, because they affect the speed of the machine upon which control depends. Of the two, a stalling rotation is the more dangerous for two reasons: (*a*) because the pressure angle is increased, which increases the resistance, and, unless the thrust of the screw is increased proportionally, the speed is decreased. This is always dangerous and many accidents have been due to stalling. (*b*) If the angular velocity of a stalling rotation is high, there will be a rapid increase of pressure per square foot on the supporting surfaces and this sudden strain may cause the machine to collapse. Several deaths have been due to this cause.

5 Two effects may be produced by a diving rotation: (*a*) the speed is always increased, and the path is downward. These of themselves are not dangerous because increased speed means better control and, provided there is sufficient altitude, the machine can be brought back to the horizontal before it strikes the earth. (*b*) If the angular velocity of the diving rotation is too high, a *downward* pressure is produced upon the supporting surfaces which may cause their collapse downward. The explanation of this is as follows:

6 In Fig. 1 AB is the chord of a surface moving to the right. Let AH be the horizon, let $\alpha = BAH =$ the pressue angle and also the angle of incidence, as we are assuming calm air. If the aviator moves the lever so that the angle of incidence is decreased and becomes $\alpha - \triangle \alpha$, the pressure angle also is decreased, hence the lift is decreased and the machine no longer maintains a horizontal course. Assume the new path to be AC and let the angle $CAH = \beta$. When $\alpha = \beta$ the new pressure angle becomes $\alpha - \triangle \alpha + \triangle \alpha = \alpha$ and the pressure is still upward. The danger in a diving rotation depends upon the fact that the maximum value of β per unit time is fixed for a machine of a given weight and speed, so that if, in that time, $\triangle \alpha > \alpha + \beta$, the condition shown in Fig. 2 where AC is the path, is obtained and the pressure is downward.

7 Too rapid a diving rotation has caused the downward collapse of machines and the deaths of some aviators.

8 The rotations described above are caused by the movement of the horizontal rudder and it is evident that the angular velocity of these rotations will determine the magnitude of the danger which follows.

9 I have shown that, by moving the horizontal rudder at the proper speed, it is possible to reduce the angle of incidence without altering the pressure angle. This is safe because with a constant pressure angle there can be no sudden change in the pressure per square foot. I have thus far assumed that there is no wind or that the wind is steady and horizontal. As a matter of fact this condition rarely exists and it is necessary to consider

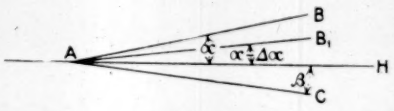

FIG. 1

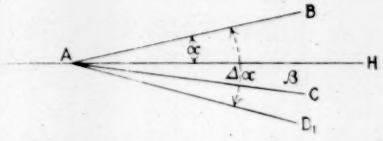

FIG. 2

FIG. 1 DIAGRAM SHOWING HOW THE ANGLE OF INCIDENCE CAN BE REDUCED WITHOUT ALTERING THE PRESSURE ANGLE

FIG. 2 DIAGRAM SHOWING THAT WHEN THE ANGLE OF INCIDENCE IS REDUCED TOO QUICKLY THE PRESSURE ANGLE BECOMES NEGATIVE AND THE AIR STRIKES ON TOP OF THE SURFACE

the problem when flying in a natural wind. Any change in the direction of the natural wind alters the value of the pressure angle and the position of the point of application of the resultant pressure upon a supporting surface is a function of the pressure angle. With any curved surface, such as is always used in flying, when the pressure angle is decreased from 90 deg. to about 15 deg. the center of pressure moves forward, but from 15 deg. to 0 deg. it moves backward. This phenomenon is called the reversal of the center of pressure and it makes the flying machine, as at present designed, inherently unstable. A graphical representation of the movemenut of the center of pressure on a curved surface is shown in Fig. 3. In Figs. 3 and 6, *a* represents upward and *b* downward pressure.

10 At all flying angles (+ 3 deg. to + 10 deg.) the center of pressure moves forward as the pressure angle increases. Let us see what effect this has on stability:

11 In Fig. 4, let AB represent a surface held against a wind AC; then BAC = the pressure angle. Let the point O represent the center of pressure and the center of gravity. Suppose the wind to change to the position AD; then it is evident the pressure angle has been increased by the amount CAD. Let $BAC = 5$ deg. and $CAD = 3$ deg., then the new pressure angle will be 8 deg. Fig. 3 shows that when the pressure angle increases from 3 deg. to 8 deg. the center of pressure moves forward. Let the new center of pressure be at

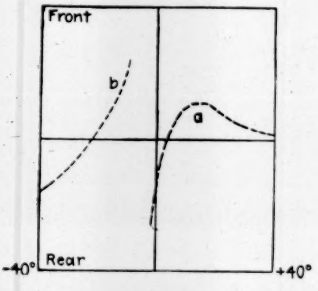

FIG. 3 DIAGRAM OF THE MOVEMENT OF THE CENTER OF PRESSURE UPON A CURVED SURFACE SHOWING THAT AS THE PRESSURE ANGLE INCREASES FROM 0 DEG. TO 15 DEG. THE CENTER OF PRESSURE MOVES FORWARD, WHICH PRODUCES AN UPSETTING COUPLE

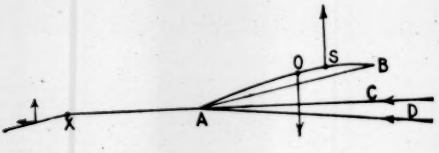

FIG. 4 DIAGRAM SHOWING HOW THE DANGEROUS STALLING COUPLE IS OFFSET BY THE HORIZONTAL RUDDER

S. This produces a couple tending to cause a stalling rotation, and is just the reverse of what is desired, because, if any rotation takes place about the lateral axis, and some rotation is bound to take place when the wind changes its direction, that rotation should be such as to keep the pressure angle constant, which is generally the important thing in flying. Therefore, it is evident that the couple shown in Fig. 4 is an upsetting couple and that safety requires that it should be offset by some other couple. As a matter of fact this other couple is introduced by the horizontal rudder shown at X, but the point to note is

that this couple must be introduced by the pilot and that the machine, of itself, has no stability. I consider this to be a fundamental error in the design of all flying machines which must be remedied if there is to be any large future for the art.

12 All naval architects know that when a boat tips to the left the center of buoyancy must move to the left in order that the meta-center shall be above the center of gravity. Now, referring to Fig. 4, it will be seen that the change from AC to AD is equivalent to a contra-clockwise rotation about the center of gravity (that is, to the left), letting AC remain stationary, yet the center of pressure moves to the right. The offsetting of this false couple with the horizontal rudder would find its analogy in naval architecture if, when a boat tips to the left, it were neces-

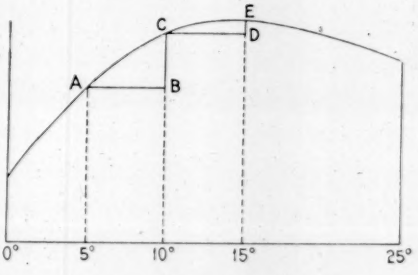

FIG. 5 DIAGRAM OF THE LIFT GRAPH OF A CURVED SURFACE SHOWING THAT THE VALUE OF A TANGENT TO THE GRAPH DECREASES AS THE PRESSURE ANGLE INCREASES

sary to use a pontoon on the left side to keep it from turning over. If such a construction were necessary with boats, their use would be very much restricted; yet just such a construction is necessary with the present type of flying machine. That we fly as well as we do is not due to the design of the machine, but to the skill of the pilot.

13 So far as longitudinal stability is concerned, the most superficial study of the subject should teach that a machine must be designed so that when the pressure angle increases, the center of pressure will move backward constantly through all angles from 0 deg. to 90 deg. Let us see if we can solve the problem theoretically:

14 Fig. 5 represents the lift graph of a curved surface and in its fundamental characteristics it is typical of all lift graphs. The

abscissae represent the pressure angles and the ordinates represent the lift. A glance shows the important fact that the value of the tangent to the curve decreases as the angle increases. From 5 deg. to 10 deg. the increased lift is BC, while from 10 deg. to 15 deg. it is only DE. Therefore if one surface is placed behind another in a position such that the pressure angle of the front surface is always greater than that of the rear surface, then, when the angle of the *system* is increased, the center of pressure will move back as it should and we shall have a righting couple. Such a system, known as the converging tandem system, has been tested in the laboratory of M. Eiffel and the graph of the movement of its center of pressure is shown in Fig. 6.

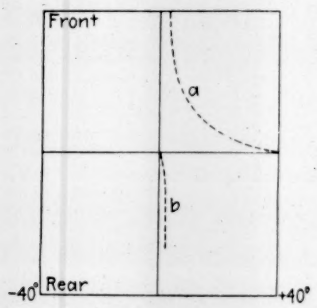

FIG. 6 DIAGRAM OF THE MOVEMENT OF THE CENTER OF PRESSURE ON A CONVERGING TANDEM SYSTEM, SHOWING THAT THE COUPLE INTRODUCED IS A RIGHTING COUPLE

15 Note that, at small angles, the pressure jumps back to the rear edge, but it is downward so it still produces a righting couple. The method of testing for the center of pressure consists simply in setting the surface with its span vertical on a vertical axis in a current of air, and noting the pressure angle for each particular position of the axis relative to the front edge. With the ordinary curved surface, designed as monoplane or biplane, it is impossible to obtain stability at flying angles, whereas, with the converging tandem, the system can be moved away in either direction and, when free, will return to its original position under the influence of the air. In fact it acts exactly as does a weathervane. I have tested this in speeds up to 120 miles per hour. This system, then, so far as laboratory tests show, has inherent longitudinal stability, and its practical value is sup-

ported to some extent by the fact that all the long flights with models (over 2700 ft.) are made with it. At the time of writing, it has not been given a thorough test with a full sized machine so far as I know. It must not be confused with the tandem system Langley used. In this the pressure angles of the two surfaces were equal, hence it could not have longitudinal stability, as a study of Fig. 5 will prove.

16 Considering the question of the tandem further, with the present type of machine the false movement of the center of pressure necessitates an offsetting couple (produced by the horizontal rudder) and it follows that the machine must be sensitive to the rudder, hence its moment inertia about the lateral axis must not be above a certain limit. This means, however, that, if the machine is sensitive to the rudder, it is also sensitive to gusts. On the other hand, by the very nature of its structure, the tandem will always have a larger moment inertia about the lateral axis than a monoplane or biplane of equal area and weight; hence it will not be as sensitive to its rudder, neither will it be as sensitive to gusts. Criticism has been made of the large moment inertia of the tandem, but I fail to see why this characteristic is a dangerous one, since a righting couple is always introduced when the machine is disturbed. It is evident that the course of a tandem cannot be changed as quickly as that of a monoplane, but that is a point in its favor, not against it. The course of a touring car cannot be changed as quickly as that of a bicycle, yet no one claims that a bicycle is safer than a touring car.

17 There is another point in favor of the tandem. As air leaves the rear of a surface it is moving downward. This constitutes the slip stream of the surface. Somewhere back of the surface the air rises, seeking its normal position. It is possible that a position may be found in the wake where a surface can be placed to great advantage. At the present time, with monoplanes and biplanes, all the energy delivered to the air by the front surface is wasted. It is evident that few naval architects have studied aeronautics else more thought would have been given to using this wake. The tandem offers a chance to utilize this wake.

LATERAL STABILITY

18 Lateral stability has to do with rotations about the longitudinal axis. The resultant pressure on a surface is a function

of two independent variables. The pressure varies as the square of the speed and as less than the first power of the pressure angle. It is evident therefore that a change of speed has more effect on pressure than a change of angle. The relation of the pressure to the speed is constant for all kinds of surfaces, but its relation to the angle depends upon the shape of a cross-section of the surface used.

19 I will consider first the forces involved in producing a rotation about the longitudinal axis when flying in calm air, which is called a banking rotation. Suppose a machine to be flying straight and the vertical rudder is turned. This causes a rotation about the vertical axis and makes the tip speeds unequal. The change of tip speeds produces a change of pressures and causes a banking rotation. The machine banks in the same direction it turns; thus, if the rudder is set for a left turn the right wing moves faster than the left wing, the former rises and the latter falls so that the machine banks to the left. This is as it should be. If we had assumed in the first place that the machine was off its balance laterally, then a turn towards the high side, by retarding the high wing and accelerating the low wing, will bring the machine back to the horizontal position.

20 As in all other phenomena connected with flight the time element is a factor which determines the nature of the result which follows the use of the rudder. In a machine in which the moment inertia about the longitudinal axis is greater than the moment inertia about the vertical axis, too quick a rotation about the vertical axis will spin the machine without giving it time to bank and skidding will result, since without a bank, the direction of the path of a machine cannot be changed. On the other hand, the proper angular velocity of rotation about the vertical axis will produce the proper bank for the turn it is desired to make. The vertical rudder therefore plays an important part in all rotations about the longitudinal axis.

21 However there are other means for producing banking rotations. The one most commonly used was first reduced to practice by the Wright Brothers, and it consists of warping the supporting surface in a manner such that the pressure angles at the tips are unequal, the larger angle being at the tip which must be raised. Practically all machines flying today use this system, either as the Wrights use it or in a modified form. Practically all machines increase the pressure angle of the tip which must be

raised. If the supporting surface is not warped, auxiliary surfaces called ailerons are used to produce substantially the same result. While machines using this system fly, and fly well in the hands of good pilots, there is a fundamental error in this system which increases the danger of flying. Since the pressure varies as the square of the speed it is evident that to change the angle of a tip, which must be raised, in such a way as to reduce its speed will produce a result the opposite of what is desired unless some offset is introduced. When the pressure angle, say of the right tip is increased, the resistance offered by that tip is increased and its speed decreased, hence the machine starts to turn to the right and a banking couple to the right is produced. However this is just the reverse of what is wanted, because in-

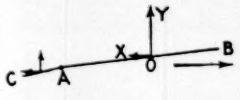

FIG. 7

FIG. 8

FIG. 7 FORCE DIAGRAM OF A SURFACE AND AN AILERON SET AT A POSITIVE ANGLE SHOWING THAT THE *H* AND *V* COMPONENTS PRODUCE OPPOSITE EFFECTS

FIG. 8 FORCE DIAGRAM OF A SURFACE AND AN AILERON SET AT A NEGATIVE ANGLE SHOWING THAT THE *H* AND *V* COMPONENTS COÖPERATE TO PRODUCE SIMILAR EFFECTS

creasing the pressure angle was for the purpose of raising the right tip, while a banking rotation to the right will lower that tip. It is at this point that the rudder comes into play as an offset. The rudder is turned so as to prevent any rotation about the vertical axis and hence keeps the tip speeds equal, then the difference in the pressure angles produces a banking rotation to the left, which is what is wanted. With lateral stability as with longitudinal stability flying machines are designed so badly that a dangerous couple which should not exist has to be offset with another couple controlled by the pilot. The horizontal rudder has to be used to prevent stalling and the vertical rudder has to be used to prevent the disastrous effect of the warp alone. In the Wright machine, but only in the Wright, there is a mechanical connection between rudder and warp which relieves the strain on the pilot somewhat, nevertheless the design is radically wrong because it introduces a force which is not only useless but

which is positively dangerous. It is a very simple matter to design a system in which all the forces introduced can be used and Figs. 7 and 8 will explain how this can be done.

22 In Fig. 7 let AB represent a surface moving in the direction of the arrow and let OY and OX be respectively the V and H components of the pressure upon it. Suppose it is desired to raise this surface and for this purpose an aileron, which hitherto has been out of action, is moved down to the position AC. The component pressures on this are represented by the arrows. It is evident that if the speed is kept constant the surface will rise, but as the H component of the system is increased, the speed will decrease and instead of rising the surface may fall. The problem can be worked out as follows: Compute the horsepower necessary to support the surface alone with a given pressure angle and speed, the weight of the surface being known. This horsepower is to be constant. Move the aileron down to a certain angle and compute the horsepower necessary to support both surface and aileron at the original speed. If the aileron is given a positive angle, which must be done if the surface is to be raised, it will be found that more horsepower is needed; but we have assumed the horsepower to be constant, therefore it is evident the speed of the system must decrease. Since when there is no change in the angle, horsepower varies as the cube of the speed, it is an easy matter to compute the new speed. Knowing the new speed and the pressure angles of the surface and aileron the new lift of the system can be found.

23 These computations are based upon coefficients determined in a laboratory. With Eiffel's coefficients I can show that if the power remains constant and the pressure angle increases, for the first 2 or 3 deg. there is a slight increase of lift, afterwards the lift drops rapidly. This is what I call the fallacy of the positive angle. Within the last six months foreign students are beginning to understand it and as able an expert as Captain Duchene has written upon the subject. He says:

> It would seem that in aeroplane construction designers have refused to employ anything but controls acting *differentially;* that is, every time one particular organ of control had to fulfil a definite function it has been so designed, with the greatest ingenuity, as to fulfil precisely the opposite function as well. Sometimes even—and especially with the warp on large-span machines—this opposite effect preponderates and must be overcome by bringing another organ into play. But in any case, the differential

action of the controlling organ compels the designer to make its dimensions unnecessarily large.

Whether with warp or ailerons, the effect aimed at is to raise one wing-tip by increasing its lift, and to lower the other wing-tip by decreasing its lift.

But if we inspect Eiffel's curves for lift or drift we see that for every increase in the angle of incidence, though the lift may increase (which is the effect aimed at by warping), the drift also increases, which is a parasitical effect, for the ensuing drag on the raised wing-tip tends to produce a turning movement in the opposite direction to that which should follow on the inclination of the wing produced by altering the lift on the wing-tips.

The warp therefore acts differentially, so that in large-span machines where the drag is considerable, its detrimental effect has to be counterbalanced by an opposite effect obtained through the use of the rudder.

The necessity for operating the warp in conjunction with the rudder therefore only results from a defect in the method of warping which, as known at present, is a barbarous method.[1]

24 The question now is what can we substitute for the positive angle? From Fig. 7 it is seen that an upward V component and a backward H component must always antagonize each other. As the H component must always be backward suppose a downward V component is used. Instead of pulling an aileron *down* suppose it is pulled *up* on the opposite tip, i.e., on the tip that is to be lowered.

25 Fig. 8 shows what will happen: The V component of the aileron is downward and the increased H component of the system, by retarding the tip, also lowers it. The components work in harmony and no offset is needed. Can anything be simpler or safer?

26 I have shown that the converging tandem should produce inherent longitudinal stability. Any disturbance about the lateral axis will introduce a righting couple which will set up oscillations, which will gradually die out. The magnitude and perodicity of these oscillations will depend upon the strength and direction of the gust, the moments of the pressure on the surfaces about the center of gravity and the moment inertia of the whole machine about the lateral axis. One important feature about these oscillations is their effect upon the lift of the surfaces. As the bow rises the lift will increase, and as the bow falls the lift will decrease. This will result in an undulating flight without however materially alterating the altitude.

27 In calm air it is possible to obtain inherent lateral sta-

[1]Aeronautics (English), July 1913.

bility by what is called a dihedral angle. If a line joining the tips of a surface passes above the center of the surface the surface is said to have a dihedral angle. If such a surface is rotated about its longitudinal axis one side of it will become horizontal and the other will move away from the horizontal, hence the difference in the V components on the two sides will cause the surface to move back to its original position through a series of oscillations. But these oscillations differ from those about the lateral axis since at no stage of these oscillations is the lift greater than it was when flying horizontally, while at some stages the lift is very much less, hence such oscillations will cause a great loss of altitude. Of course these oscillations are damped out in time and so a dihedral angle will prevent a machine from turning over, but this does not mean that to depend upon a dihedral angle is safe. Less than 1 per cent of aviation deaths are due to overturning. The largest number of deaths are due to striking the earth at too high a speed with the machine at such an angle that it does not strike *first* on its landing gear. Assuming a loss of manual control in a machine having a dihedral angle there is the highest probability that when the machine finally strikes the ground it will not land on its wheels, and to land in any other way means a smash and possible death to the pilot. Unless they have made a careful study of this subject, few people realize that, for safety in landing, it is necessary not only that the machine should be right side up but that its longitudinal and lateral axis should be within at least 2 or 3 deg. of the horizontal. The dihedral angle alone will never be able to guarantee this condition of safety.

28 Owing to the fact that any system of inherent stability must set up a series of oscillations, and as oscillations about the longitudinal axis always mean a loss of altitude which is a very dangerous thing, it appears to be impossible to obtain a safe system of inherent lateral stability. Certainly, the chances of finding it are less than they are of finding a safe system of inherent longitudinal stability. The converging tandem offers a possible solution of the problem of inherent longitudinal stability because no matter how long the machine oscillates under the combined action of the gust and the righting couple there is no danger of its losing altitude as there always is from oscillations about the longitudinal axis.

29 There is another kind of stability, however, which offers a

chance of making flying safer. This is automatic stability. If a mechanism could be invented which would move the control levers in the proper way and at the proper time, the pilot would be relieved of much strain and flying would be made safer. Since stability is wholly unrelated to translations of the machine and connected only with rotations about one or more axis, it follows that the control mechanism must be sensitive to rotations but not sensitive to translations. This eliminates the pendulum as a possibility. Contrary to belief the pendulum is unsuited for this purpose, because a translation of its point of support sets up oscillations which are not wanted. Mercury cups or other moving fluids are also unsuited for the same reason, i.e., they are sensitive to translations of the machine. The gyroscope is unsuited because it tends to maintain a fixed plane, whereas in practical flying the longitudinal and lateral axis must often take different planes at different times. Moreover, there is one characteristic of the gyroscope which is very bad. It cannot act except it is revolving. It is dangerous to depend for stability upon a mechanism which has to be in motion.

30 This summer at Chicago I examined a mechanism, the details of which I am not at liberty to disclose, which appears to be able to guide a flying machine independently of manual control. It is sensitive to rotations but not to translation, the couple introduced is a static couple, the force employed cannot fail to act and the distributing mechanism is so simple it is hard to see how it can get out of order. To move the levers, power is taken from the engine, the mechanism simply controlling how far and when the levers shall be moved. The system is auxiliary to manual control and can be cut out whenever the pilot so desires.

31 Inherent stability for flying machines is hard to obtain. The problem is different from that of a boat, because a boat is immersed in two fluids of different densities, while a flying machine is immersed in one fluid. There is no reason, however, why stability cannot be greater than it is in existing machines: two fundamental errors in design should be changed now, namely, the use of the horizontal rudder to prevent stalling and the use of the vertical rudder as an offset to the warp. For some years I have been convinced that these offsets are evidences of bad design, and apparently other students are coming around to my opinion.

32 It is 22 years since Langley published his valuable work on plane surfaces, yet, although the Wrights were the first to fly,

no research work worthy of the name has been done in this country and the only scientific progress the world has made had its start in Eiffel's laboratory researches.

33 I hope that this paper will awaken an interest in the theory of aeronautics to the end that engineers will give some of their time and, where able, some of their money, to any proposition which will put aeronautics upon a firm foundation in this country. It is to be regretted that, although aviation had its start here through the work of Langley, Chanute and the Wright Brothers, we are at the present time far behind France so far as real scientific progress is concerned.

34 What is necessary is several well equipped laboratories and able men who can devote all their time to research work. Before aviation is placed upon a firm foundation a correct theory of design must be worked out, and this can be accomplished only by thorough research work in the laboratory.

THE ART OF ENAMELING OR THE COATING OF STEEL AND IRON WITH GLASS

By Raymond F. Nailler

ABSTRACT OF PAPER

The paper opens with a brief history of the art of enameling, going back to the very early stages of its development; the present-day methods of enameling are then outlined. The composition of enamels is treated, showing the chemical actions which take place, the substitution of materials, the theory and practice of what cause the enamel or glass to adhere to steel or cast iron, and the various ingredients used to produce various colors in enamel or glass.

The recent development of enameling larger apparatus and a brief description of this art close the paper.

THE ART OF ENAMELING OR THE COATING OF STEEL AND IRON WITH GLASS

By Raymond F. Nailler, Elyria, Ohio

Member of the Society

Before going into the technology of this art, it will be interesting to note its history in a general way: where it originated and how it was brought down to the present age. The covering of burnt earthenware, porcelain and some metals with a crude enamel took place about the same time as the discovery of glass. Since the time when man first enameled metals, especially gold, silver and copper, he began to be interested in the problems connected with artistic enamels.

2 Colored enamel earthenware has been found in the ruins of Thebes. In the ruins of a great many other ancient cities of Egypt enameled or glass-covered brickwork has also been discovered. That the Egyptians knew how to adorn silver vessels with enamel pictures has been recorded by Pliny the Elder. From Egypt the enameling arts were transferred to Greece and thence to Rome, and some historians maintain that the enameling art came to Italy by Arabia, Spain and the Balearic Islands, and through Roman expeditions the art passed into England, France and Germany. In the museum at Oxford is an enameled ornament which was found in Somerset, the inscription of which dates to the time of Alfred the Great.

3 As a particular example of early art enameling let us briefly consider the so-called enamel painting. Two periods may here be distinguished: The first, known as the "old Limoges style," was characteristic of the time of Francis I (1515-1547). The enamel plate, generally made of copper, was covered with a dark enamel coat. After firing, figures, weaker or stronger accord-

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ing to the relief desired, were put on with white enamel and the impression of bas-relief thereby brought about. The second period of enamel painting began a generation after the first. This is the so-called period of the "miniature style," which was introduced about the middle of the sixteenth century. It was brought to an extraordinary state of perfection by Jean Petitot (1607-1690).

4 As the demand for the artistic enamels of antiquity and the Middle Ages increased, so the enameling of gold and silver gave way to that of copper vessels, and from this point, if we except the somewhat scattered industry of miniature painting on boxes, such as jewel cases, etc., the absolute decline of the enameling art is to be recorded.

5 In the nineteeth century, which it must be conceded brought one of the most important of Europe's social, educational, and technical revolutions, the enameling business came into great significance and eventually became the enameling industry of today. The commercial introduction of iron and steel made it possible to re-apply the half-forgotten art of enameling iron utensils.

6 In the history of the development of enameling or the modern industrial technology of iron enameling, we are able to speak of two periods with regard to the iron materials employed, viz., of the original enameling of cast iron exclusively and the later application to sheet metal.

7 Before considering the technology of any particular type of enamel it will be well to consider the meaning conveyed by the word enamel. If the present-day enamel be briefly and scientifically designated as a boro-sodium-potassium-aluminum silicate generally colored by metallic oxides, then the following definition given by Popelin deserves to be quoted since this, notwithstanding its length, may be described as most clear and comprehensive: "Enamel is a glass fusible at a low temperature and usually compounded of borates and silicates. This mixture originally colorless combines with the greatest ease with metallic oxides under the influence of a pyrotechnic operation, thereby acquiring various colors according to the nature of the oxide which the enameler can vary at will."

8 The uses to which the general class of ceramic compounds known as enamel is put are varied. We have noted that enamels are used for artistic or decorative purposes. Parallel to this

use we may classify a far more important type of enamels as "commercial," and it is with this type of enamel that we are to concern ourselves. The first class of commercial enamels is that used on cast iron and this classification includes the field of sanitary equipment together with various forms of cast-iron kettles and similar pieces of engineering apparatus. The enameling of sheet steel may be taken as the second class of commercial enamels. The most familiar division of this class includes cooking ware enamels. The various forms of so-called agate and granite ware as well as the single colors are made by enameling sheet-steel forms. The second class of steel enameling which has more recently come into industrial importance is the manufacture of enamel signs, and the third field includes the manufacture of heavy equipment for large scale food preparation, the dairy industries, and general chemical operations. The apparatus in this last case may take the form of tanks, kettles, evaporators, pipes, etc.

9 In outlining the technology of the enameling industries as a whole we may include to a certain extent all of the above commercial classes, certain details of which are varied to meet the conditions of cast iron and various types of sheet-steel products. The outline following, however, is characteristic of all classes.

10 The first step to be considered is the preparation of the enamel. The purity of the raw materials to be used in the compounding of an enamel must be certain and in the cases where materials can vary in strength the actual analysis of the substance must be known to assure proper results. In many cases the secret of an enamel lies in its formula and so the compounding of the batches is very carefully guarded, only a person of responsibility having charge of weighing the ingredients. The various materials are generally kept in bins which are numbered, the person in charge of the mixing having the formula stated in terms of these numbers. The properly weighed batch is thoroughly mixed, this being accomplished either by shoveling carefully on a specially prepared floor or by mechanical mixing by means of a rotating agitator.

11 The product of the mixing room is taken to the smelter in which the various ingredients are fused together in the form of a mass having the characteristics of glass. The furnace in which this operation takes place is a special reverberatory furnace similar to that used in the puddling process in the manu-

facture of wrought iron. The coal is placed on a grate at the front end of the furnace and the burning gases pass over the bridge wall, strike the roof, and are deflected against the batch. The first change which may be noted in the smelter is the driving off of the water from the borax which produces a swelling up of the mixture. The fluxing materials then melt down and as the heating is continued, dissolve the more refractory constituents. If the batch is properly stirred and the temperature of the furnace carefully regulated, the product of this operation is a clear transparent glass containing no particles of unfused material. When a test of the fusion comes up to this standard the furnace is tapped and as the liquid glass flows into a tank of cold water it is broken up by the chilling action of the water, the result being a thorough granulation of the product. The material thus prepared is known as "frit."

12 The next step in the preparation of the enamel involves grinding the frit to a fineness at which it can be applied to the surface of the metal. The mill here used is the ordinary pebble mill lined with porcelain brick and containing very hard flint pebbles. In the cast-iron industry two general processes are used, the dry and the wet, in sheet-steel work only the wet. In the first case the frit is ground dry, the pulverized enamel being sprinkled on the hot cast-iron piece. In the wet process a certain percentage of pure white clay is placed in the mill with the frit and a certain definite amount of distilled water. When certain grades of enamel are to be manufactured various compounds are also added here to aid in producing desired colors, gloss and opacity. An instance of this is the addition of tin oxide in the mill in the production of white cooking utensil enamels. The fineness to which the enamel is ground depends upon the particular use to which it is to be put, manner of application and other factors. In the wet process the essential feature is that it be fine enough to remain suspended in the water assisted by the clay and the so-called "vehicles" mentioned below.

13 At this point we may well take up the consideration of the construction and preparation of the material to which the enamel is to be applied. Cast-iron enamels are applied to castings of the desired shape. In the case of cooking ware enamels the shapes are constructed by pressing and spinning. For heavier equipment such as jacketed kettles the apparatus is con-

structed by riveting or preferably welding the sheet steel in the desired form. The selection of the steel with a view to its chemical analysis is of prime importance. A reliable specification reads as follows:

Sulphur	below	0.040
Phosphorus	below	0.030
Manganese	about	0.40
Silicon	about	0.010
Carbon	about	0.10

The steel must necessarily be free from laminations or other mechanical imperfections.

14 With reference to the construction of heavy apparatus the sheet ranges from $\frac{3}{16}$ in. to $\frac{3}{8}$ in. in thickness. Two general methods of construction are in use: The first involves the formation of unit sections fitted with flanges. These sections are enameled separately, bolted together at the flanges and the desired apparatus so constructed. The preferable practice, however, is to construct the apparatus in one piece by means of autogenous welding, thus avoiding the use of gaskets or other packing materials in erection. That the enamel may properly adhere to the surface of the metal, the latter must be free from dirt or scale and in the case of welded joints the welds must have a preliminary grinding to reduce the roughness. The entire surface of the apparatus is then cleaned by pickling or sand blasting, the latter process being altogether used in cleaning large apparatus. As the crude ware leaves the sand blast it has a roughened, clean, metallic surface and is in the proper condition to receive the enamel.

15 Before applying the enamel to the metallic surface it is prepared by a process known as "setting up." This involves the addition of certain chemicals to the enamel as taken from the mill, the function of this addition being to assist the clay in holding the enamel particles in suspension. Substances so added are termed "vehicles." At this stage the enamel must be diluted with distilled water to the proper consistency for application.

16 In applying the enamel to the metallic surface three general methods are in use: The first, applicable to small pieces only, is known as "dipping," the piece being dipped into the enamel, the excess of which is shaken off leaving a thin coating

on the metal. The second method, known as "slushing," involves pouring the prepared enamel over the surface and allowing it to drain. The third method, which is the principal one used on larger apparatus, involves spraying the finely ground enamel on the metallic surface by means of the compressed air atomizer.

17 Preliminary to the consideration of firing the enamel we may well review the types of furnaces in use. The first type is known as the muffle furnace and involves the use of a large fire clay oven externally heated by means of coal, gas or other fuel. The apparatus to be fired is placed on suitable supports in this muffle. The other type of furnace is known as the direct fire furnace, in which the heat from the fire is taken up by the walls of the firing chamber and radiated to the apparatus placed within the chamber on suitable racks. This general class of furnace has two special divisions in that, on the one hand, the piece is rotated within the furnace, while on the other hand, the piece is allowed to remain stationary. For small work the muffle is in general use, but for the production of large apparatus the direct-fire furnace is necessary. At first thought, the muffle furnace may be considered to have an advantage in that the products of combustion together with the dust from the fire cannot come in contact with the enamel. Again it may seem that a more even heat can be realized in the muffle. On the other hand, with a properly designed direct-fire furnace in which the combustion is complete before the gases reach the firing chamber no trouble is experienced due to their presence or to the dust from the fire. The use of natural gas further does away with this latter possibility.

18 In a furnace for firing smaller ware the charging mechanism is a fairly simple matter, it being necessary merely to place the material in the furnace by means of a small fork operated by hand or mechanically. But in the manufacture of engineering apparatus where a single piece may weigh 3000 to 4000 lb. it is necessary to have a large mechanical charging machine on which the piece may be placed outside the furnace, the arm of the machine then properly placing it in the furnace. The general design of such a machine suggests the charger used in open-hearth practice. The apparatus to which the enamel has been properly applied is placed in the furnace which is maintained at the proper temperature. This temperature varies with the nature of the enamel and in cases of high silicon acid-proof

enamels reaches in the neighborhood of 2500 deg. fahr. The control of the burning is made possible by the changes which occur in appearance of the enameled surface as fusion takes place. At first the fine particles of enamel begin to fuse together and a general blister condition exists, giving the surface a very dull appearance. But as the enamel matures, this dull appearance gives way to a bright glass, which when properly developed over the entire surface is indication that the piece should be withdrawn from the furnace. The time required to burn a piece properly depends upon the temperature of furnace, thickness of metal and nature of enamel.

19 Nothing has been said so far as to the composition of the enamel or of the number of coats applied. In general there are two kinds of enamel, known as ground coats and cover coats. The former serves as a bond between the enamel and the steel, and the latter serves to build up the body of the enamel and presents the finished surface. In the ground coat color is no object. Its composition is such as to render it adherent and strong. In addition to the ordinary components cobalt oxide seems to be essential to the production of adherence. The explanation of this is debatable. The cover coat is the one which forms the major part of the enamel and if definite color, opacity, etc., are objects the necessary ingredients for their production are introduced here, assisted by mill additions as noted above. In the manufacture of acid-proof enamel, the cover coat is essentially a high silicate and must be free from any metallic oxides, such as oxide of tin, lead, iron, etc. The piece to be enameled receives one ground coat which is burnt well into the steel at a high temperature. The cover coats may be one or two in number for ordinary enameling, but should be at least triple for acid-proof work. In the production of acid-proof apparatus, the use of a ground coat or cover coat is now eliminated and the same material which in place of being an enamel, as commonly termed, is in reality a boro-silicon glass and without the use of any metallic oxides in its compounding.

20 It is very interesting to note the chemical changes which take place in the various stages of the production of the finished enamel. Avoiding so far as possible deep technicality, they may be summed up as below. As a starting point let us select a cover coat formula used in the production of a "dark blue" cooking ware enamel.

Feldspar, lb.	120	Saltpeter, lb.	7
Quartz, lb.	72	Oxide of Cobalt, lb.	7½
Borax, lb.	80	Oxide of Manganese, lb.	1
Cryolite, lb.	30	Clay in the mill, per cent	4

First considering the behavior of each of the constituents when heated we can later clearly note the general reactions which take place when they are fused together in the smelter.

21 Feldspar varies greatly in composition but as commonly used in enamels is an aluminum-sodium-potassium silicate of approximately the following analysis:

	Per Cent
Silica (SiO_2)	70
Alumina (Al_2O_3)	17
Soda (Na_2O)	7
Potash (K_2O)	6

On smelting, none of these constituents vaporize so we may consider them all later in the smelting of the above enamel.

22 Quartz introduced as a pure glass sand contains practically 100 per cent silica (SiO_2).

23 Borax is chemically known as sodium tetra-borate and in the crystalline form as used has a certain definite amount of so-called water of crystallization which must be taken into account. The formula $Na_2B_4O_7.10H_2O$ is resolved into $Na_2O.2B_2O_3.10H_2O$. Without going through the calculations assume this to correspond approximately to the following analysis:

	Per Cent
Water (H_2O)	16.0
Boric Oxide (B_2O_3)	37.0
Soda (Na_2O)	47.0

Of these the last two do not vaporize on smelting, but the water is evaporated, hence 100 lb. of borax smelts to 84 lb. of the remaining oxides.

24 Cryolite is a double fluoride of sodium and aluminum, the formula of which may be written Na_3AlF_6. When smelted, the sodium and aluminum appear as oxides and from 100 lb. of cryolite we realize about 24 lb. of alumina (Al_2O_3), 44 lb. of soda (Na_2O) and 54 lb. of fluorine (F_2). There is some dispute as to whether or not the fluorine is vaporized. In the present discussion it matters not, hence we shall consider that the third of these three compounds is lost in smelting.

25 Saltpeter is potassium nitrate KNO_3. When heated under

the conditions of smelting it may be considered to break up into potash (K_2O) and nitrogen pentoxide (N_2O_5). The reaction is $2KNO_3 = K_2O + N_2O_5$. By calculations based on this reaction 100 lb. of saltpeter yields about 47 lb. of potash (K_2O) and 53 lb. of the nitrogen oxide (N_2O_5). The latter may be considered as completely vaporized.

26 Oxide of cobalt (CoO) may be taken as non-volatile and pure. The same may be assumed for the oxide of manganese (MnO_2).

27 To sum up the above as an outline of the reactions taking place in the smelter

120 Lb. Feldspar	gives	$120 \times 0.70 = 84.0$ Lb.	Silica (SiO_2)
		$120 \times 0.17 = 20.4$ Lb.	Alumina (Al_2O_3)
		$120 \times 0.07 = 8.4$ Lb.	Soda (Na_2O)
		$120 \times 0.06 = 7.2$ Lb.	Potash (K_2O)
72 Lb. Quartz	gives	$72 \times 1.00 = 72.0$ Lb.	Silica (SiO_2)
80 Lb. Borax	gives	$80 \times 0.16 = 12.8$ Lb.	Water (H_2O) (Vaporized)
		$80 \times 0.37 = 29.6$ Lb.	Boric Oxide (B_2O_3)
		$80 \times 0.47 = 37.6$ Lb.	Soda (Na_2O)
30 Lb. Cryolite	gives	$30 \times 0.44 = 13.2$ Lb.	Soda (Na_2O)
		$30 \times 0.24 = 7.2$ Lb.	Alumina (Al_2O_3)
		$30 \times 0.54 = 16.2$ Lb.	Fluorine (F_2) (Vaporized)
7 Lb. Saltpeter	gives	$7 \times 0.47 = 3.3$ Lb.	Potash (K_2O)
		$7 \times 0.53 = 3.7$ Lb.	Nitrogen Oxide (N_2O_5) (Vaporized)
7½ Lb. Cobalt Oxide	gives	$7\frac{1}{2} \times 1.00 = 7\frac{1}{2}$ Lb.	Cobalt Oxide (CoO)
1 Lb. Manganese Oxide	gives	$1 \times 1.00 = 1$ Lb.	Manganese Oxide (MnO_2).

We shall consider that of the above the H_2O, N_2O_5 and F_2 are vaporized. This leaves for the constituents of the frit (totals of the above)

	Lb.	Per Cent
Silica (SiO_2)	156.0	53.5
Alumina (Al_2O_3)	27.6	9.5
Soda (Na_2O)	59.2	20.3
Potash (K_2O)	10.5	3.6
Boric Oxide (B_2O_3)	29.6	10.2
Cobalt Oxide (CoO)	7.5	2.6
Manganese Oxide (MnO_2)	1.0	0.3
Total	291.4	

The loss, theoretically, on smelting is plainly $317.5 - 291.4 = 26.1$ lb. or $\frac{26.1}{317.5} = 8.2$ per cent. (317.5 lb. = original weight of batch).

28 The mill additions together with the agents used in "setting up" are fused with the frit as the piece is fired. By a process similar to the above, we could compute the final composition of the enamel by taking these additions into consideration. The actual amount of material added in this case, however, is so slight and of such a nature that the change resulting therefrom is not sufficient to affect materially the composition of the enamel. When an addition of about 12 per cent tin oxide accompanies the clay, a very significant change in the composition of the enamel is produced.

29 It may be well to note in passing some of the means by which various colors are produced in the enameling industries. It will be impossible to enter into great detail without taking too much time, but the mention of certain compounds in connection with the colors produced by their use will serve our purpose.

30 The production of a good white enamel either for cast-iron or sheet-steel work may be said to depend, at the present time, upon the use of tin oxide. Great have been the efforts to substitute less expensive substances, such as compounds of antimony and lead. But an antimony white which looks good alone is plainly seen to be off-color when compared with a good tin oxide white.

31 Going to the other extreme of color, black, we encounter difficulties. There is any number of formulae for black enamels, but when the results are closely compared we find that the colors range widely through brown blacks, blue blacks, purple blacks, etc. Certain compounds of manganese and iron used together give a color approaching black. Other formulae call for the combined use of oxides of manganese, cobalt and copper. Again we find oxide of nickel added to the above three oxides.

32 A color much seen in enamels is blue and the use of cobalt is very satisfactory in the production of this color in various grades of intensity. Manganese alone produces purples and violets and in combination with cobalt gives various shades of purple-blue.

33 Green enamels are chiefly produced by the use of chromium oxide and copper oxide, while in some cases a mixture of copper and cobalt oxide is used.

34 Reds of various shades are produced by the use of red oxide of iron. In connection with it we find that tin oxide aids greatly in giving opacity and bringing out the color. In the

production of brown enamels we may use ferrous chromate. Various yellows are produced by salts of cadmium, chromium and uranium.

35 The more delicate shades of rose and purple are produced by the use of gold compounds. So-called "pink-rose" is used in the manufacture of certain artistic enamels. Perhaps the best known gold compound used in enamel coloring is "purple of Casius." The exact composition of this product is a question. It is made by the combined use of auric, stannous, and stannic chlorides. The color produced is also commonly called "purple of Casius."

36 Before drawing this paper to a close, attention is invited to a general consideration of the future of the enameling industry. Neglecting art enameling and sign making we come to the field of steel enamels. So far as the cooking ware industry is concerned, the field is practically constant. Granting that the demand for that class of article is increasing, as the public becomes accustomed and educated to its use, there is an opposing tendency in the rapidly increasing use of aluminum ware. Exactly how these and other factors now balance would be difficult to ascertain. But aluminum is a metal and its metallic properties cannot be denied. Under certain conditions it is attacked by various substances used in the culinary arts and a contamination of the preparation is inevitable. No doubt the time will come when a high silica enamel known to be free from tin and other poisonous compounds will enter the cooking ware field. The government is becoming more and more careful in protecting the public from foods of injurious nature and it is not too much to expect that soon it will establish more rigid restrictions relative to the ingredients entering into the manufacture of apparatus in which food is to be prepared. At such a time an enamel coming up to requirement will be free from injurious compounds and will come into great demand.

37 In the preparation of foods on a factory scale, we find an enormous and constantly increasing demand for larger pieces of enameled steel apparatus in the form of pans, kettles, tanks, pipe, etc. There are many lines of pressure being brought to bear both by the government and public opinion which lead to the conclusion that the increasing demand for this style of apparatus is without limit. Canning and preserving factories and dairy establishments have found a large use for copper and tin in the construction of containers, vacuum pans, etc. The acids

of fruit juices and vegetable pulps have a very marked action on these metals and the resulting contamination of the product is known to be of danger to the consumer. The use of an enamel containing tin, lead or other metallic oxides is but the first step in the right direction. The presence of these metallic oxides in the enamel renders it corrodible and contamination results. The solution is the use of an acid-proof enamel free from all such poisonous substances. In the milk industry a similar line of reasoning applies. Further compare the ease of maintainig sanitary conditions in a one-piece enamel lined unit with the trouble experienced in the use of a metal container or even an enameled article made up of composite parts between which are gaskets.

38 Another consideration relates to the preparation of chemicals later used in food products, for instance, baking powder. Many operations connected with the manufacture of such products have been carried on in lead or other metallic pans and the resulting contamination has given no end of trouble. Acid-proof enamel is rapidly solving this problem also.

39 Finally consider the chemical manufacturing processes now carried on in apparatus of lead, wood and earthenware, necessitated by the corrosive actions of the liquors and gases involved. This includes the pharmaceutical field which alone is a matter of great importance. In all these and many other lines the use of acid-proof enameled apparatus is rapidly finding and filling a great demand.

40 Not only does steel apparatus meet the demands of modern industries, but in case a cheaper product is desired and at the same time a heavier construction is permissible, acid-proof cast-iron apparatus has its field. The possibility for size and variety of construction is, of course, more limited than in the case of sheet-steel apparatus.

41 In view of these considerations and many others which these have called to mind, we cannot but conclude that the use of enameled apparatus has just begun and with this extension of the long known art of metal enameling, a field of great industrial possibilities both for manufacturer and user has been opened. We may not be criticised as being over optimistic when we predict that in their ultimate stage of development the enamel industries will be ranked among the greatest of commercial activities. At such a degree of development the enameling industry will in no way deserve classification among the lost arts.

A NEW CENTRIFUGAL PUMP WITH HELICOIDAL IMPELLER

By C. V. Kerr

ABSTRACT OF PAPER

The reason given for seeking a new type of pump, in view of the fact that standard forms of centrifugal pumps have been so improved in recent years that very creditable performance has been reached, is based on the characteristic behavior of the small steam turbine commonly used to drive pumps for the class of service in view, namely, circulating pumps for condensers in power plants and irrigation or drainage pumps where large volumes of water at low lift are to be handled. Calibration curves for different types and sizes of steam turbines are given showing the relation of power and speed at different steam pressures. Examples illustrating the economy of the high-speed as compared with the low-speed unit are given.

A brief treatment of the mathematical theory of the new type of impeller is presented covering especially features of helical and conoidal surfaces under rotation.

Detailed description of the 8-in. experimental pump is presented with methods and results of tests. Efficiency curves, the capacity of the impeller for centering itself and for maintaining high-suction lift, capacity for handling air leak in suction piping, and comparative behavior of ring-oiling and water-lubricated bearings are fully described.

Tests also on a 30-in. pump are fully described and a method of deriving and locating the parabola of highest efficiency at different speeds for such pumps is explained. In addition to the characteristic curves for this large pump the curve of pressures at different speeds under shut-off, together with the pressure ratio curve is given.

On the basis of these tests a series of pumps was designed and a typical pump in cross-section is shown and described.

The location of the parabola of highest efficiency having been explained, a method of locating it with respect to the friction curve in surface condenser work, which is also a parabola, is presented with mathematical treatment in brief and applied to the case of a pump designed for such service.

A NEW CENTRIFUGAL PUMP WITH HELICOIDAL IMPELLER

By C. V. Kerr, Wellsville, N. Y.

Member of the Society

THE PURPOSE

The progress made in recent years in the performance of centrifugal pumps is gratifying. Pressure pumps suitable for boiler feeding or waterworks service are showing efficiencies as high as 80 per cent at fairly high speeds, and low-pressure pumps at low speeds have shown efficiencies above 60 per cent and even as high as 86 per cent.

2 With many firms building centrifugal pumps with performance like those noted, of excellent construction and running qualities, the use of seeking a new type of pump may be questioned. The reason is chiefly in the characteristic behavior of the small steam turbine which is commonly used as the driving power for the class of service in view, as circulating pumps for condensers in power plants or drainage pumps with low lift and variable speed turbine or motor drive.

3 This small steam turbine is a product of recent years, and it has been making a place for itself rapidly and creditably. Its strong points for power plant service in driving auxiliaries are reliability, clean exhaust-steam for feedwater heating, and small operating expense for attendance and repairs. As part of the power plant, the steam turbine by its oil-free exhaust returns to the boiler through the feedwater heater possibly 90 per cent of the total heat of the steam supplied to it. On the other hand, an electric motor as a driver of auxiliaries, besides being less reliable, represents a waste of probably 85 per cent of the total heat in the steam passing through the main power units into the condensing water.

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4 The turbine operating with steam at high velocity is essentially a high-speed machine. If the turbine were held still, the same total weight of steam would flow through the nozzles as if it were running at the most economical or best speed. To illustrate the relation between power developed by a turbine and rotative speed the following diagrams are used.

5 Fig. 1 shows the calibration of a 12-in. Kerr steam turbine, which was used for tests on pumps within its limits of power and speed. The gage pressure noted on the curves represents

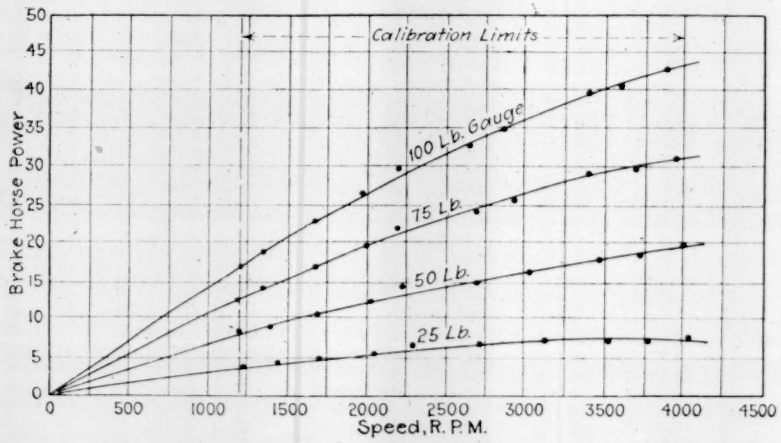

FIG. 1 CALIBRATION OF 12-IN. KERR STEAM TURBINE

the pressure between the governor valve and the nozzles of the first stage. The power developed was measured by a special form of high-speed prony brake, and the back pressure in exhaust was maintained at atmospheric pressure. The limits of speed within which these observations of power and speed were taken are shown on the diagram. If the turbine were to be held fast by this prony brake, obviously no power would be delivered; the calibration curves are therefore properly extended to the origin or coördinates. It should be understood at once that these curves represent a constant steam flow in pounds of steam per hour, although the power delivered varies with the speed. Note that with a constant pressure of 100 lb. gage, the turbine at 1200 r.p.m. delivers 17 b.h.p., while at 4000 r.p.m. 43 h.p. are developed. The same total steam in pounds per hour flows through the turbine in both cases, and

the water rate is inversely proportional to the power developed. Note also that with the lower steam pressures, the power speed curves are flatter than with the higher steam pressures. This means that the turbine reaches its best speed, or speed of greatest economy, sooner with low-pressure steam. This turbine with steam at 150 lb. gage would probably find its best speed between 5000 and 6000 r.p.m.

6 In the same way Fig. 2 shows the calibration of a 36-in.

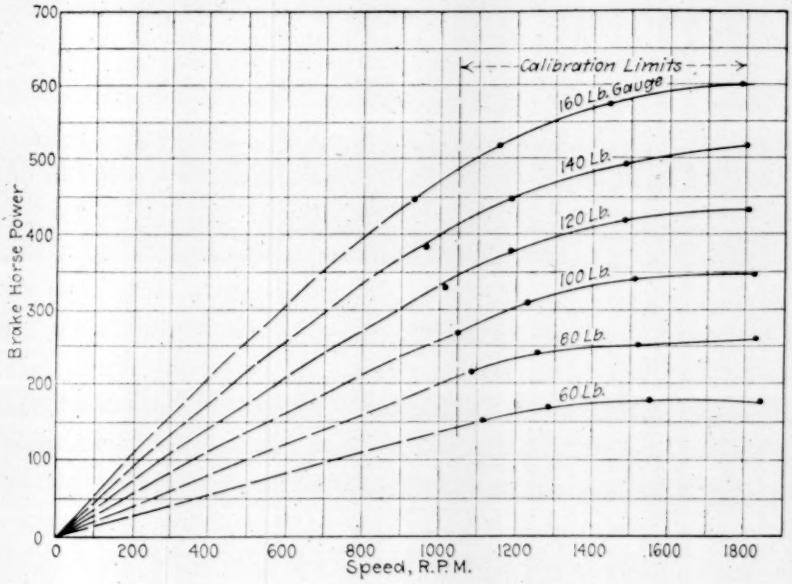

FIG. 2 CALIBRATION OF 36-IN. KERR STEAM TURBINE

Kerr steam turbine used in testing the 30-in. pump to be described later. For the ordinary working range of steam pressure, the best speed of this turbine is found near 1800 r.p.m. On the curve of 160 lb. gage, it will be seen that this turbine at 600 r.p.m. develops about 320 b.h.p., while with the same total steam in pounds per hour at 1800 r.p.m. it develops fully 600 b.h.p. Suppose such a turbine to be driving a pump at 1500 r.p.m. and delivering water at such rate that 350 b.h.p. are required at a pump efficiency of 65 per cent. The curves show that a pressure of 102 lb. on the first stage nozzles would produce the required power. Assume a standard centrifugal pump of 80 per cent efficiency to be doing the same work at a speed

of 600 r.p.m. and requiring 275 b.h.p. The steam pressure necessary is 146 lb. and as the steam flow and first cost are proportional to pressure; the cost of running the high-speed pump will be only 70 per cent of the low.

7 That such behavior is quite characteristic of the steam turbine is indicated by Fig. 3, which shows the calibration of a type *BC* Terry turbine running condensing. Although this turbine with 100 lb. gage develops only 100 b.h.p., its best speed is found to be 2400 r.p.m. This turbine was built to run a 16-in. pump

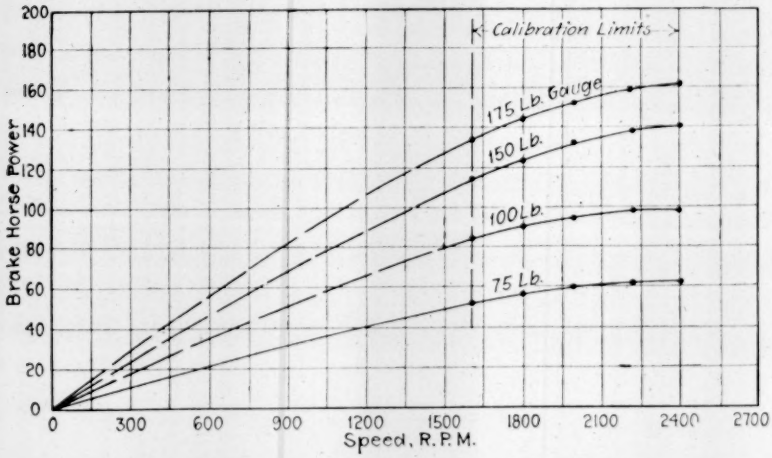

FIG. 3 CALIBRATION OF CONDENSING TERRY TURBINE

at 2200 r.p.m. at a total head of 35 ft. It is important that both initial and final pressure of the turbine shall be maintained on test of a driven machine the same as when under calibration. If these points are observed, then it is probable that the turbine thus calibrated is for practical purposes as accurate and reliable as a calibrated electric motor, which is much more commonly used for such purposes.

8 The small steam turbine, like the large one, is not worth much without something to drive. In the early days in the development of the small steam turbine, much trouble and some painful experiences were met in securing suitable centrifugal pumps for the steam turbine to drive. The pumps available had been built for use with steam engines or electric motors so far as possible. They were adapted to their speeds either by direct connection or by belt drive. Further, the class of service for

which they were used called generally for cheap construction as well as reliability in service. In the case of high-pressure centrifugal pumps, the conditions of speed were at first good, and improvement has been made both in raising speed and improving the construction. But it is with the low-head or low-pressure centrifugal pumps used for hot-well and condenser service in power plants that the old trouble of low speed and to some extent of cheap construction still obtains.

9 To show something of what a change in speed may mean in power plant service with steam turbine drive, consider a few instances of centrifugal pumps of best construction and performance running at different speeds. Take the case of a pump referred to above; its efficiency of 71 per cent at 1145 r.p.m. is creditable, but if compared with a high-speed pump running at 3200 r.p.m. for the same capacity of 1200 gal. per minute at a head of 50 ft., it will, when driven by the same size of steam turbine, have a total steam consumption in pounds per hour of some 960 lb. greater. Assuming an evaporation by the boilers of 8 lb. steam per lb. of coal under actual conditions, with coal at $3 per ton, the excess of steam by the slow-speed pump will call for 180 tons of coal per year, which at $3 per ton amounts to $540 per year. Again, take the pump with the record efficiency of 86 per cent at 860 r.p.m., which is a typical pump for barometric condenser service. If driven by a steam turbine, it would have a total steam consumption per hour 1060 lb. greater than that of a smaller turbine which could be used to drive a pump at 2000 r.p.m. and handle the same quantity of water, 5500 gal. per minute against 45 ft. head, with an efficiency of only 62 per cent. With the same evaporation and cost of coal, the excess coal required would be worth $600 per year of 3000 hours. Even more extreme cases may be cited. Suppose a pump of 25,000 gal. a minute capacity for condenser circulating water to run at 600 r.p.m. against a total head of 45 ft. As compared with a pump of the same capacity at 1500 r.p.m., the slow-speed pump with an efficiency of 72 per cent as against the high-speed pump at 65 per cent would require with the same size of turbine some 9900 lb. more steam per hour. This under the foregoing assumption means $5570 more per year for coal.

10 It may be urged that the exhaust steam from the turbine can be used for feedwater heating, that it makes no difference what the actual water rate of the turbine or its total steam

per hour may be. But the purpose of a typical power plant is to generate and sell current for power and lighting service. Suppose we consider taking steam from the main turbine at atmospheric pressure for feedwater heating as against using the excess steam from the more wasteful steam turbine auxiliary. In the case of the large circulating pump, the excess steam means some 300 b.h.p. extra. If this steam were used in a main turbine, probably 1 kw-hr. could be developed for each b.h.p. between initial and atmospheric steam pressures. If these 300 kw-hr. are sold at 3 cents per kw. during 3000 hours per year, the total revenue for the station by this saving alone will amount to $27,000 per year. In many power stations, however, the growing use of high vacuum means more power spent on auxiliaries, also better economy in the main generating units. It may very well happen, therefore, that this excess steam must be wasted as atmospheric exhaust. Adding the cost of extra coal and the value of the extra current which could be developed by the main unit, we have $32,570 per year as the difference in the revenue of the station as between wasting and using this extra steam.

11 For such reasons as those stated or indicated in what precedes, and perhaps for other reasons also, the development of a new type of centrifugal pump, especially for low-pressure volume service, which will run at speeds to bring out the best economy of the steam turbine, and therefore of the unit, has been attempted. As part of that purpose a machine which is of the same superior construction and running qualities as the steam turbine has been kept in mind.

THEORY

12 To produce an axial movement of the water a helical or, more properly, a helicoidal surface is assumed. The intersection of the helicoidal surface with a cylindrical surface, concentric with the axis, is the ordinary helix or screw thread. The helicoidal surface is generated by a line, in this case at right angles to the axis, which moves along the axis at a uniform rate while revolving around it also at a uniform rate. The distance traveled along the axis in one turn is called the pitch. The angle made by the direction of the helix on the developed surface of a cylinder of any assumed diameter with the plane of rotation, which is normal to the axis, is called the helix angle.

13 The pitch of the helix is constant for all radial distances from the axis. Let s be the pitch of the helix, a the axial acceleration, and t the time of one revolution. Then $s = \frac{1}{2}at^2$, from which $a = \frac{2s}{t^2}$ and $ma = \frac{2ms}{t^2}$. Hence the impulse imparted is constant for all radial points. In a practical way, this means that water will be picked up at all points from the hub of an impeller to the rim with the same impulse or acceleration and that the axial flow per sq. in. will be constant.

14 To produce the radial flow required to transfer the water

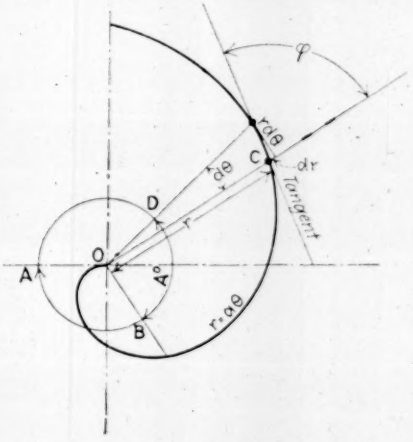

FIG. 4 ARCHIMEDES SPIRAL

from the impeller of the pump to the casing a radial impulse is necessary. In Fig. 4 is shown the arc of an Archimedes spiral which has made part of a turn about the origin at O. The equation of this spiral is $r = a\theta$, in which r is the distance from the center O to any point c on the arc; θ is the arc of unit radius subtending the spiral, or measuring the angle swept by the radius r; and a is a constant. From this equation $dr = ad\theta$; hence the radial acceleration is constant since mdr is constant for uniform rotation. Again the tangent of the angle ϕ between a line passing through two consecutive points of the spiral and the radius to one of these points is $\tan \phi = \frac{rd\theta}{dr}$. But $dr = ad\theta$ and therefore tangent $\phi = \frac{rd\theta}{ad\theta} = \frac{r}{a}$. If $r = 0$, then $\tan \phi = 0$,

which means that the spiral starts radially from the center 0. If $r = \infty$, then $\tan \phi = \dfrac{r}{a} = \infty$. Hence the spiral finally crosses the radius at right angles.

15 Again from the equation of the spiral, $\theta = \dfrac{r}{a}$. Comparing this with $\tan \phi = \dfrac{r}{a}$, the result is $\theta = \tan \phi$. This is true for $\theta = 0$, since $\tan 0^\circ = 0$; also, when finally the spiral is at right

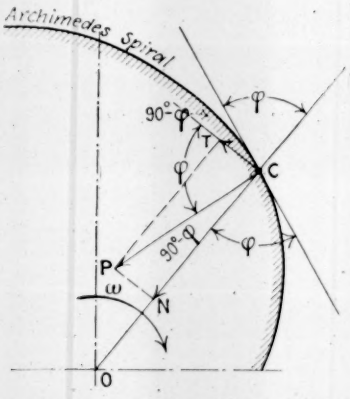

FIG. 5 PRESSURE ON SPIRAL SURFACE

angles to the radius, since $\tan 90^\circ = \infty$. Hence $\tan \phi = \theta$ is the measure of the angle swept by the radius r.

16 To find the angle in degrees of the spiral between assumed radii, $dr = ad\theta$, from which $d\theta = \dfrac{dr}{a} = \dfrac{dr \tan \phi}{r}$, in which dr may have any value from 0 to r, and the value in degrees will be

$$A^\circ = 180^\circ \times \frac{d\theta}{\pi} = \frac{180^\circ}{\pi} \times \frac{dr}{r} \times \tan\phi = 57.3^\circ \times \frac{dr}{r} \times \tan\phi$$

17 Suppose a portion of a spiral surface to be rotating with angular velocity ω about a spiral center O, as indicated in Fig. 5, and that a heavy particle at c is free to move under forces acting upon it. The radial or centrifugal force will be

$$N = m\omega^2 r$$

and the tangential force will be

$$T = m\frac{dv}{dt} = m\omega\frac{dr}{dt}$$

since $v = \omega r$ and $dv = \omega dr$. Also from $r = a\theta$, $dr = ad\theta$. Therefore

$$T = m\omega a \frac{d\theta}{dt} = m\omega^2 a$$

since

$$\frac{d\theta}{dt} = \omega$$

Hence

$$\frac{N}{T} = \frac{m\omega^2 r}{m\omega^2 a} = \frac{r}{a} = \tan\phi$$

Now the force T is at right angles to radius r and to force N, therefore the resultant force P is at right angles or normal to the tangent and the spiral, which is the path of the moving particle c. This must be true since angle ϕ of Fig. 5 is a complement of the

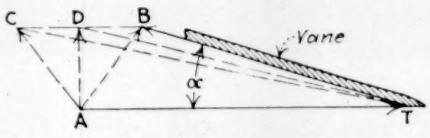

FIG. 6 ENTRANCE VELOCITIES

angle $90° - \phi$. This means that the pressure of the mass c is normal to the spiral surface, regardless of the amount of mass or speed of rotation.

18 Now combine the properties of the helix and spiral by allowing the spiral to follow or lie in the helicoidal surface. As the spiral moves out from the axis uniformly with the rate of rotation, it follows that the spiral in space, if projected on a plane through the axis, will follow the surface of a cone; while, if projected on the plane of rotation, it will appear in its usual spiral form. By this combination, water may be picked up by a helicoidal surface and compelled to flow axially over a conical surface, which gives the fluid a radial component of velocity. Now combine a pair of right and left hand impellers, embodying such helicoidal and spiral properties, and the double suction impeller itself is complete as to fundamental ideas.

19 The trouble found with high-speed propellers used on turbine shafts in the early days of marine propulsion was cavitation, which seems to mean that at a certain rate of rotation the water is unable to follow the wheel or to be picked up by it. Conditions which make for the highest practicable speed of rotation

are indicated by Fig. 6. If the desired quantity of water flows into the impeller with an axial velocity AD and the impeller rotates with a rim velocity AT, the relative velocity will be DT. If the water is directed into the impeller against the direction of rotation, as at AC, the relative velocity will be increased to CT. If directed into the impeller in the direction of rotation, as at AB, then this relative velocity is reduced to BT, and at the same time the angle of the helix is increased. For such reasons the suction inlet to the pumps, which naturally must be closed, is made in volute or spiral form directing the water into the impeller in the direction of rotation.

20 The diagram of discharge conditions at the rim of the

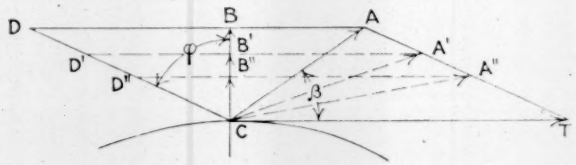

FIG. 7 DISCHARGE CONDITIONS

impeller is indicated in Fig. 7. It will be noted that, as the radial velocity of discharge CB decreases, the absolute direction of discharge CA makes a smaller and smaller angle with the tangent CT to the impeller. This change of discharge angle makes trouble with the diffusion vanes commonly used in centrifugal pumps and causes loss of efficiency through eddies. As the pumps in view are to be of the low-pressure volume type, it is possible to dispense with the diffusion vane and use instead an annular nozzle surrounding the impeller through which the fluid may be discharged and reduced in velocity to terminal pressure, as in the case of water flowing through the delivery tube of an injector. In Fig. 8 are shown the essential parts of a pump drawn to scale from a design in which the spiral or volute suction passage is indicated, also the annular nozzle surrounding the helicoidal impeller. The direction of rotation of the impeller is indicated; and also the movement of water outward through the wheel and an assumed spiral path along which the particles of fluid pass from impeller to volute casing. By plotting distances along this spiral path against the radial distances from the axis and the axial width of the annular nozzle, the effective or actual length of nozzle is increased and indicated

by the dotted lines and arrow in the left-hand portion of the figure.

21 It was shown in connection with Fig. 5 that the pressure on a particle following a spiral path is normal to it. Since the particle discharged from the impeller through such an annular nozzle moves as a free body under the forces acting the assumption is made that such a particle will tend to follow a spiral path out into the volute. The velocity of discharge from an impeller will vary widely with speed, capacity and head, so that the actual

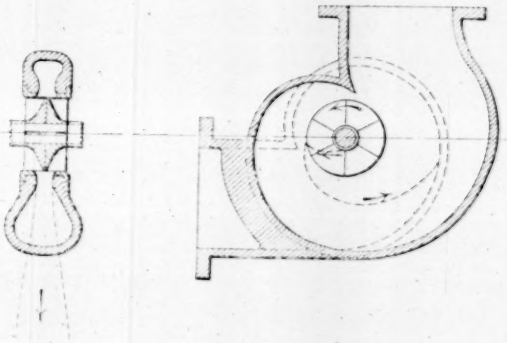

FIG. 8 VOLUTE CASING SHOWING AUTOMATIC CHANGE OF EFFECTIVE NOZZLE LENGTH

form of the annular nozzle must be a compromise. In general, as the capacity decreases, the velocity of discharge will increase, and the direction of discharge will make a smaller angle with the tangent to the impeller; but, fortunately, the smaller the angle which the direction of discharge thus makes, the longer will be the effective spiral path through the nozzle to meet the demands of the higher velocity. Thus the annular nozzle in a way automatically changes to meet more or less closely the conditions of operation.

8-IN. EXPERIMENTAL PUMP

22 The first step in the development of an idea is to fix upon the best way of carrying on the work under existing conditions, which in this case means facilities for construction and testing. For such reasons, the experimental pump shown in Fig. 9 was decided upon. In view of the avowed purpose of these pumps to work with condensers as circulating and hot-well pumps, and with steam turbines as the motive power, an 8-in. pump with a

speed of about 3000 r.p.m., a total head of from 30 ft. to 40 ft., and a maximum efficiency near 1200 gal. per min. was designed. Provision for getting the results of various combinations of impellers and annular nozzles was provided as indicated in Fig. 10 at *A* and *B*. As seen from the photograph in Fig. 9 the casing is split; and as part of the program, the flanges were planed off to reduce the nozzle width, as indicated at *A*, Fig. 10, after it had passed through the changes in form numbered 1, 2, and 3 shown

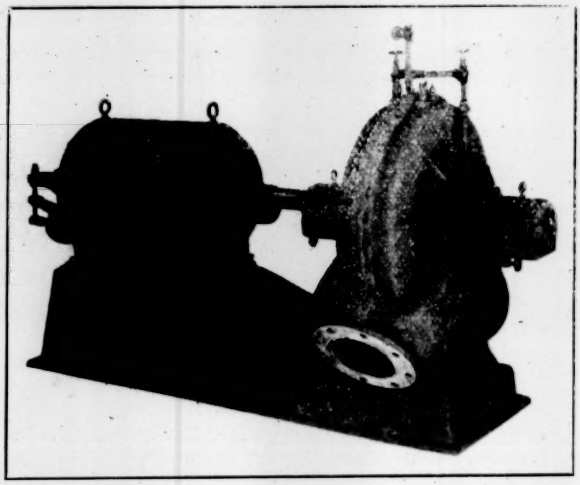

FIG. 9 EXPERIMENTAL 8-IN. PUMP

at *B*. Various forms of impellers were also used. The result of the first effort at designing, pattern-making, casting and machining was an efficiency of only 33 per cent. Indications of what changes should be made were obtained, however, and steady progress was made with some determination of what should and what should not be attempted.

23 With a double suction impeller, having six right-hand and six left-hand blades making an angle of 15 deg. with the plane of rotation and a diameter of 6½ in. for this 8-in. pump, the results in Fig. 12 were obtained. A pressure regulator of the spring-loaded type was used to control the speed of the turbine to secure constant head. For the head of 31 to 34 ft. the speed began at 3600 r.p.m. and ran down to shut-off at 2000 r.p.m. For a head of 21 to 24 ft., the speed began at 2850 r.p.m. and ran down to shut-off at 1625 r.p.m. The turbine used to drive this 8-in. pump is one whose calibration is shown in Fig. 1. The

effect on the efficiency curves of the pump itself is to make them longer and flatter than if the speed were maintained constant. The efficiency of the turbine as measured by power delivered for a given flow of steam will also be maintained higher. This brings out one of the strong points of steam turbine drive for centrifugal pumps, a variable speed easily and economically controlled.

24 The arrangement for testing this 8-in. pump is shown in Fig. 11. The concrete tank, with a capacity of 6500 gal. under

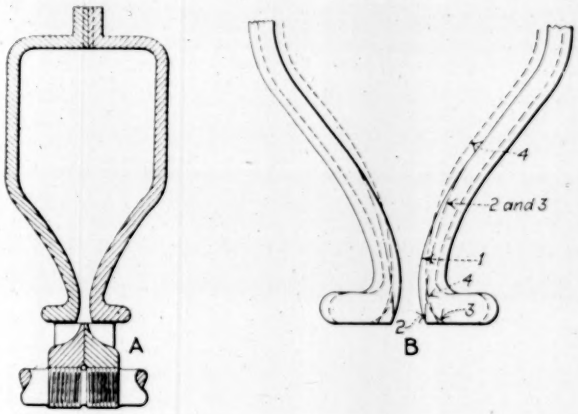

FIG. 10 ANNULAR NOZZLE OF 8-IN. PUMP

working conditions, was put down and a frame work of I-beams put over it to support flooring and apparatus. Attention is called especially to the partitions which compelled the water discharged from the nozzle to pass first over a partition, then under and finally over the last partition to the suction compartment. The water in the first or discharge compartment was churned into foam by the jet issuing from the nozzle; but when it arrived at the suction compartment in the course of operation it was quiet and to all appearance free from air, except the amount usually dissolved in water. The other features in the arrangement for testing will be readily seen, although it should be added that ½-in. pipe connections were made in order to test the capacity of the pump to handle air leaking into suction. The head gage shown is a water column with a total available range of 15 ft.

25 The pump was arranged with adjustable thrust collars in the outer bearing case. These collars could be put in the de-

sired position and were self-locking by right and left hand threads. The pump could therefore be run, either held in a fixed position by these thrust collars or left free to take its own position. This is a characteristic of the helicoidal impeller. From the construction, Fig. 10 A, it will readily be seen that if the shaft is moved to the right for instance, the discharge head on that side will increase while the capacity and suction on the opposite side will also increase. This makes a positive force

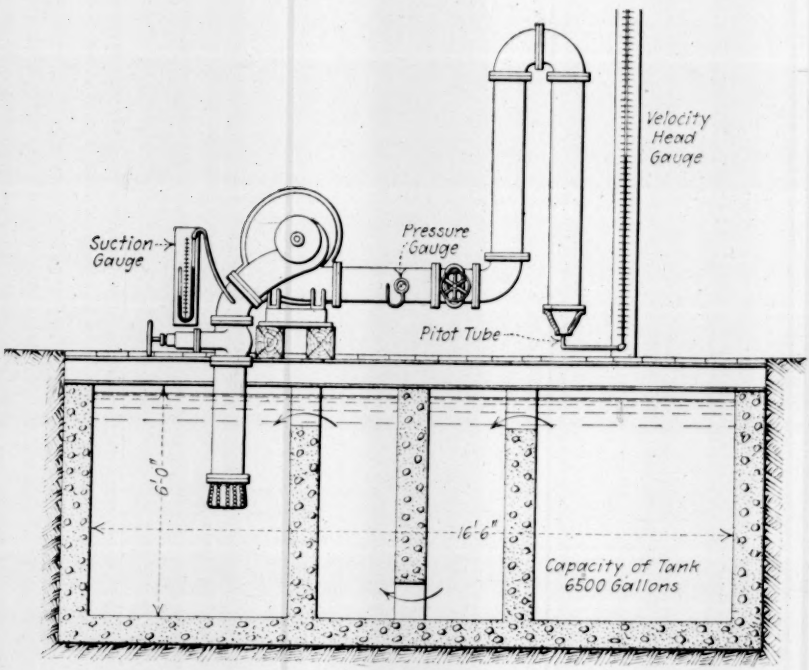

FIG. 11 TESTING TANK FOR 8-IN. PUMP

tending to bring the impeller back to central position and keep it there. This feature of the pump was repeatedly demonstrated in the course of tests made on it.

26 In testing the capacity of the pump to handle air leakage, the ½-in. pipe connections, first on one suction passage, then on both were operated. The effect of a full ½-in. opening into the suction with a suction lift of about 6 ft. was to lower the normal discharge head from 30 ft. to 20 ft. This condition would be steadily maintained until by clapping the hand over

the air opening the discharge head would instantly rise to 30 ft. and remain there. This showed that the pump could handle the large amount of air leakage, and also quickly get rid of air carried in. The ½-in. air opening in one side of the suction passage was also operated, with the result that the impeller would jump toward the opposite side of the pump in the effort to adjust the work done by the two sides of the impeller. On closing this air leak, the impeller would promptly return to the normal central position.

27 By throttling the suction the capacity of the pump to lift water was indicated. The elevation of Wellsville is about 1500

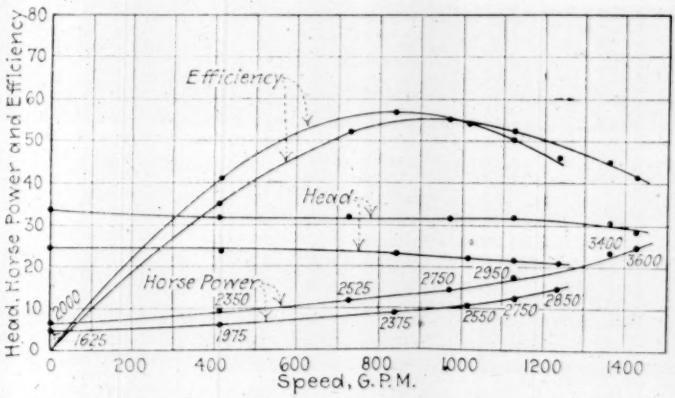

FIG. 12 TESTS ON 8-IN. PUMP AT CONSTANT HEAD AND VARIABLE SPEED

ft. above sea level, and normal barometer stands about 28.3 in. At a speed of 3700 r.p.m. and discharge head of 10 ft., with a small amount of discharge from the pump and the barometer standing at 28.05 in., the vacuum maintained by the pump, as measured by mercury column referred to the center of the shaft, on one test was 26.9 in. This corresponds to a lift of 30.4 ft. against a limit of 31.7 ft. fixed by the barometer. On another test with a different runner, with barometer standing at 28.1 in. and speed of 3500 r.p.m., the vacuum was maintained in the suction of the pump at 27.0 in., which means a lift of 30.5 ft. against a limit of 31.8 ft. fixed by the barometer. At 3040 r.p.m., the vacuum in the pump suction was maintained at 26.2 in. The highest vacuum reached could be steadily maintained at speeds as high as 4000 r.p.m.

28 The results of tests on the last combination of impeller

and nozzle are shown in Fig. 13. It is true that the results here shown do not reach as high points as those obtained from previous tests on other combinations. But it is difficult, naturally, to change impeller widths and nozzle forms successively and stop with all things in the best shape. In the meantime, also, the form of pitot tube had been changed to give more nearly correct results, and a water column had been substituted for a calibrated low-pressure gage for the pitot tube readings. The coefficient of discharge from the nozzle was assumed to be unity

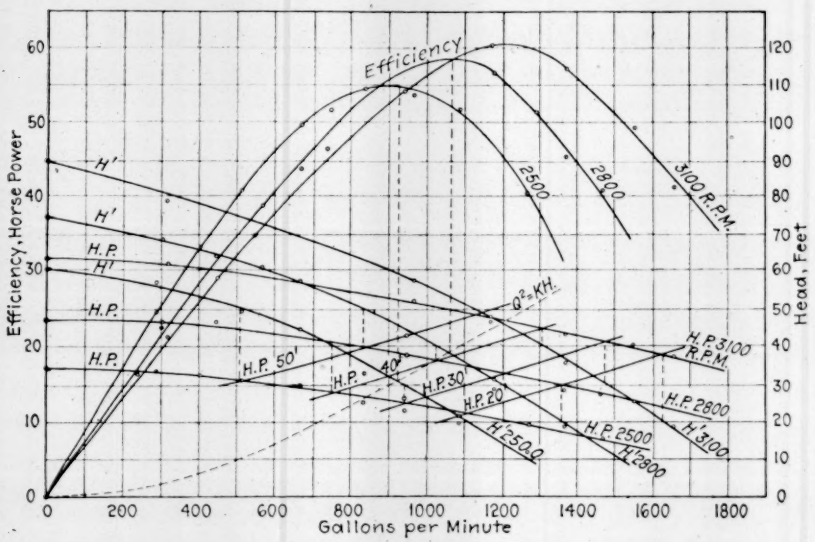

FIG. 13 FINAL RESULTS ON 8-IN. PUMP

throughout. This is justified by results obtained on nozzle calibrations at a university laboratory, and any doubts as to this point are partly balanced, at least, by the fact that the turbine driving the pump was calibrated by the builders with high-pressure steam and ran on these tests with low-pressure steam. In each case, however, the steam was passed through separators, but of different makes.

29 Most of the characteristics of the pump will readily be seen from the diagram. It would have been interesting to run another test at higher speed to see how far up in speed the efficiency of the pump would increase, but the boiler power available did not permit. The diagram indicates, among other

things, the horsepower at constant speed and also at constant head. An interesting feature in these results is brought out by the application of the equation of a parabola $Q^2 = KH$, derived as follows: $Q = AV = A\sqrt{2gH}$. Then comparing two conditions,

$$\frac{Q}{Q_1} = \frac{A\sqrt{2gH}}{A\sqrt{2gH_1}} = \frac{\sqrt{H}}{\sqrt{H_1}}$$

from which

$$Q^2 = \frac{Q^2}{H_1} \times H = K H$$

This curve will be seen to cross the head capacity curves at points

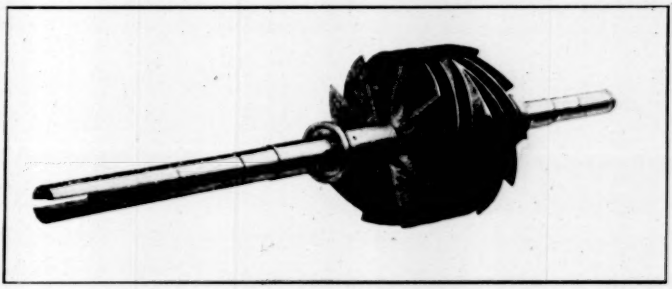

FIG. 14 IMPELLER FOR 30-IN. PUMP

which correspond to the highest efficiency reached at the different speeds. This permits prediction to be made as to head, capacity, and position of highest efficiency for other operating conditions. If in addition to this information, the envelope to these efficiency curves could be determined by more extended tests, then the actual maximum efficiency, corresponding to any speed, capacity and head within range of the pump could be fixed. This envelope will have the same general form as the efficiency curves; but it will be longer and flatter than any of them.

30 It may be interesting to add that, as a sort of side issue, metallic packing rings in the glands were tried against the ordinary lubricated flax packing. The metallic packing ran cooler and with a steady drip of water through it, which maintained the best possible working conditions, and it was found that the shaft was polished nicely under this packing. The material used was ordinary type metal made in solid V-shaped rings,

which were a close running fit on the shaft and backed by soft packing. On the other hand, the flax or hydraulic packing gave frequent trouble through need of adjustment to secure the proper compromise between heating and leaking. Also, a water lubricated bearing of type metal arranged for water from the discharge of the pump to flow through it into the suction was tried against the usual ring oiling bearing. Tests by micrometer caliper showed that the oil bearings wore the shaft more than

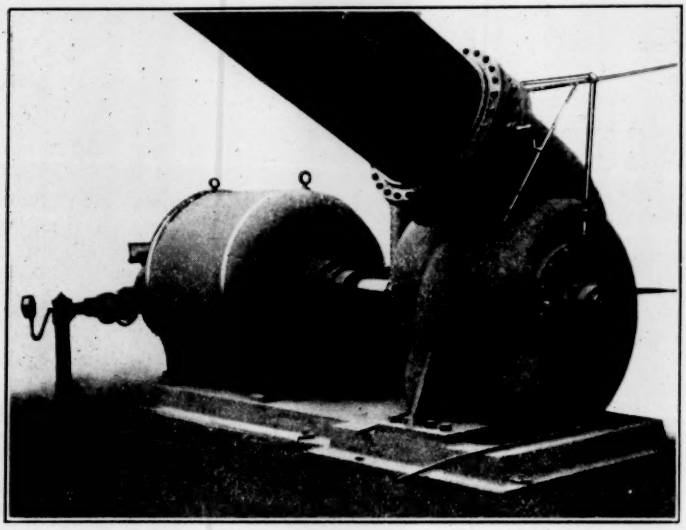

FIG. 15 THIRTY-INCH PUMP

the water bearing did. This water bearing was also arranged so that a mixture of sharp sand and water could be drawn through it into the suction passage of the pump. What such treatment would mean to a ring oiling bearing needs no comment. The water bearings stood the treatment very well.

30-INCH PUMP

31 Apparently justified by the results on the 8-in. experimental pump, the opportunity came and was accepted to build a pump of this type with the maximum capacity of 30,000 gal. at 45 ft. total head for service as a circulating pump with a large condenser. A photograph of the impeller is shown in Fig. 14. This is one of three different impellers having different numbers of blades and helix angles that were built and tested with this

pump. In this case the runner has 10 blades with the helix angle of 20 deg. The other runners were of 5 and 8 blades with helix angles of 20 deg. and 17½ deg. respectively.

32 The pump with the turbine driving it is shown in Fig. 15. The suction was vertical and the discharge 30 deg. from the horizontal. This arrangement of the pump was required by the location between the foundations of two large units which made it necessary to discharge directly into the water box of the condenser. The turbine is the one whose calibration is given in Fig. 2.

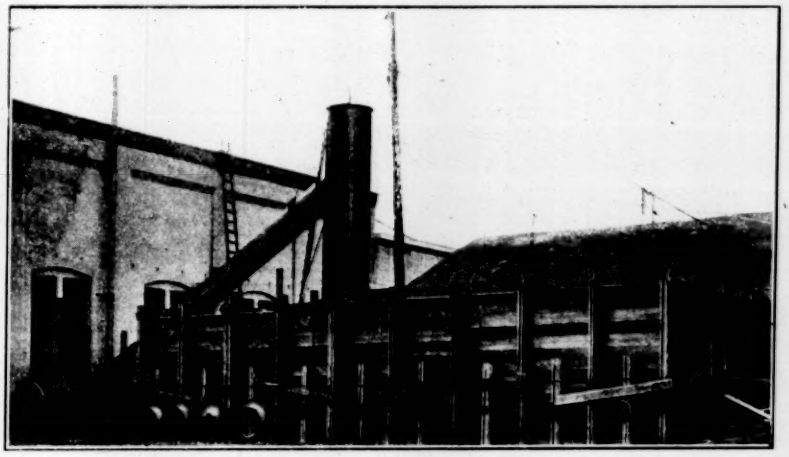

FIG. 16 TESTING TANK FOR 30-IN. PUMP

33 On account of the unusual size of this pump and the desire to learn as much as possible of its performance under different conditions, somewhat elaborate arrangements were made for testing it. A general view of the testing tank used with the turbine and pump, discharge pipe and standpipe is given by Fig. 16. The suction and discharge connections are 30 in. and the standpipe is 48 in. Two glass water columns are shown on the side of the standpipe, one for velocity head as measured by the pitot tube and the other for static pressure at the level of the nozzle. Some of the nozzles used are seen standing against the side of the testing tank.

34 The tank has a capacity under working conditions of about 45,000 gal. The water flows continually around or through the apparatus and the direction of flow in the tank is

such as to free the water of air as completely as possible. It was very effective and the water returned to the suction inlet completely freed from air, except that dissolved. This tank was designed and built after some runs had been made with a round tank holding some 12,000 gal. of water with a vertical partition in the middle to force the separation of air bubbles entrained by

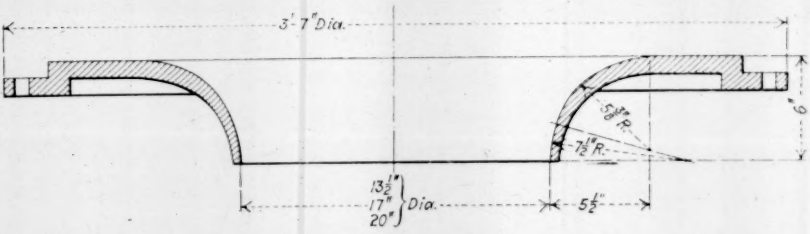

FIG. 17 NOZZLES FOR 30-IN. PUMP TEST

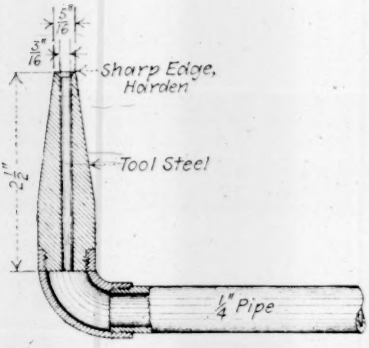

FIG. 18 FORM OF PITOT TUBE

the jet of water from the standpipe; but this amount of water furnished only a half minute's supply at full capacity and in operation the tank became so full of foam that air bubbles reached the suction pipe and apparently occasioned efficiencies incredibly high.

35 A calibrated test gage was used to determine the steam pressures at the turbine and a calibrated portable tachometer was used to determine speeds. Mercury columns were used to measure the suction and discharge heads. The suction column was connected to both sides of the pump. The discharge column was connected to the discharge pipe at four points with valves

in each branch to detect any differences in head due to the different positions; but any differences which may have existed were covered by errors of reading.

36 Three nozzles of size and form as in Fig. 17 were provided. In this case a coefficient of 0.97 was used which is thought to be sufficiently conservative. Three sizes of pitot tube of form given in Fig. 18 were used. The material was tool steel and to

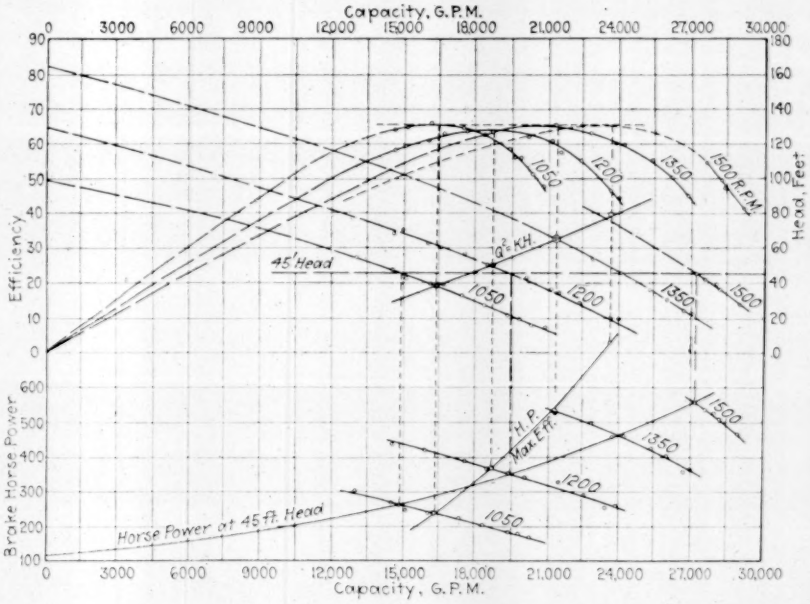

FIG. 19 RESULTS OF TESTS ON 30-IN. PUMP

prevent damage during the tests, the edge was made sharp and hardened. One smaller and one larger size than that shown were tried under similar conditions, but the $\frac{5}{16}$-in. tip was used as giving the best combination of sensitiveness and steadiness of water column. The static water column was used to check the pitot tube readings and when both were in proper working order and due allowances were made for connections and velocity heads, the agreement between the two sets of readings was very close. The advantage of the static water column as a safeguard was repeatedly shown during the tests. Occasionally a small piece of shaving or fiber of some kind would lodge on the tip of the pitot tube and lower its water column to such an extent as to

require immediate attention. This condition which might not otherwise have been noted quickly was always shown at once by comparison with the static water column.

37 The general results of reliable tests are set forth in Fig. 19. Although the boiler plant which furnished steam for these tests was of the water-tube type of 400 h.p. rating with available steam pressure of 200 lb. gage, it was found impossible to maintain maximum working power for any length of time and the contemplated range of tests was curtailed in consequence. A considerable amount of steam necessarily had to be used for shop

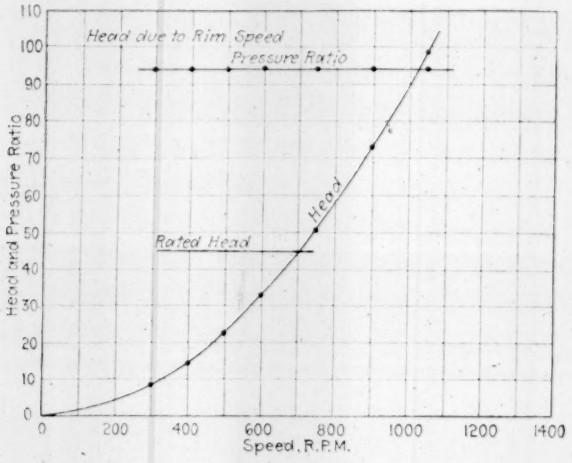

FIG. 20 SHUT-OFF PRESSURES IN 30-IN. PUMP AT VARIABLE SPEED

driving and driving auxiliaries with some routine testing at the same time. In general, the form of curves with difference in scale is very much like those obtained from the 8-in. pump. Although the head and horsepower curves rise somewhat rapidly, the horsepower for the constant rated head of 45 ft. falls rapidly with the decrease in capacity. In surface condenser service, especially where the discharge pipe as well as the suction pipe is sealed by water, the working head is largely due to friction in pipes, valves, and condenser tubes. It follows, therefore, that the head in operation will actually increase with the capacity of the pump. The power required will, therefore, vary under actual conditions even more rapidly than with constant head. This is illustrated in Fig. 19 by the line marked horsepower at maximum efficiency.

38 The efficiency actually reached is thought to be creditable, considering the fact that this is the first large pump of the type to be built and that the speed is much higher than is ordinarily attempted with centrifugal pumps under like conditions of head and capacity. The effect on the unit consisting of turbine and pump as to steam per water horsepower may be readily guessed by comparing the power developed by the turbine at speeds of 1200 and 1500 r.p.m. with that at 600 to 800 r.p.m. when combined with such efficiencies as were actually secured.

39 Some interesting information in regard to the behavior of this type of impeller at shut-off or no-discharge is conveyed by the diagram in Fig. 20. With a blank flange in the discharge pipe from the pump the turbine was run at different speeds and

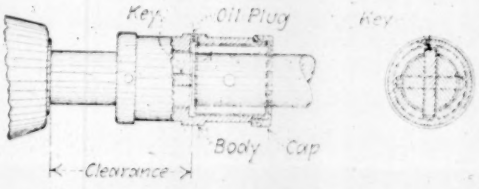

FIG. 21 LUBRICATED FLEXIBLE COUPLING

the corresponding total head in feet of water observed. The rated head of 45 ft. was obtained at 700 r.p.m. The lower pressures on this curve were obtained by the mercury column and the higher pressures by the calibrated test gage. The head due to rim speed is figured from the rim velocity of the impeller at different speeds. The pressure ratio line shows the relation between the head actually observed and that due theoretically to rim speed. To find this ratio practically constant at about 94 per cent was somewhat of a surprise.

40 The desire to make the design of the pump such that the impeller and shaft could be withdrawn endwise from the casing made it necessary to design some other form of flexible coupling than the commonly used flanged rubber bushing type. The result is shown in a general way in Fig. 21. The essentials of this coupling are a pair of flat tool-steel hardened keys set in the adjacent ends of the shafts at right angles to each other. The power is transmitted from one to the other through a crucible cast-steel body which is closed at each end by a cap which screws tight against a shoulder. Through the oil plug indicated, a

heavy lubricant is put into the coupling, such as cylinder oil or machine oil mixed with graphite. The spaces in the coupling may be filled with this oil until it begins to run out at the shaft

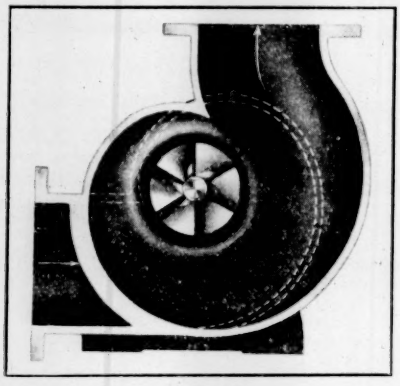

FIG. 22 TYPICAL SECTION OF PUMP CASING

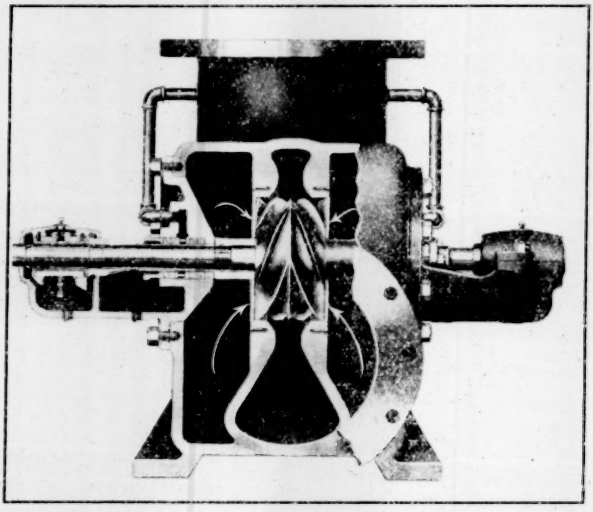

FIG. 23 SECTION OF PUMP SHOWING HELICOIDAL IMPELLER IN PLACE

level. In operation, the centrifugal force throws this oil out into the grooves where the keys are bearing on the body. The object of hardening the keys is to maintain the form of the key and to put all the wear on the soft steel body of the coupling. By re-

moving one end cap, the shaft may first be withdrawn far enough to drive out the straight pin used to hold the key in central position, which permits the removal of the key and then the impeller from the casing. To illustrate the capacity of this coupling, one size is 4 in. in diameter, weighs 15 lb. and transmits at 2400 r.p.m. a maximum of 300 h.p. with safe working stresses.

TRAITS OF THE HELICOIDAL PUMPS

41 Upon the basis of information obtained from such tests as the foregoing, and from observation of running qualities, a

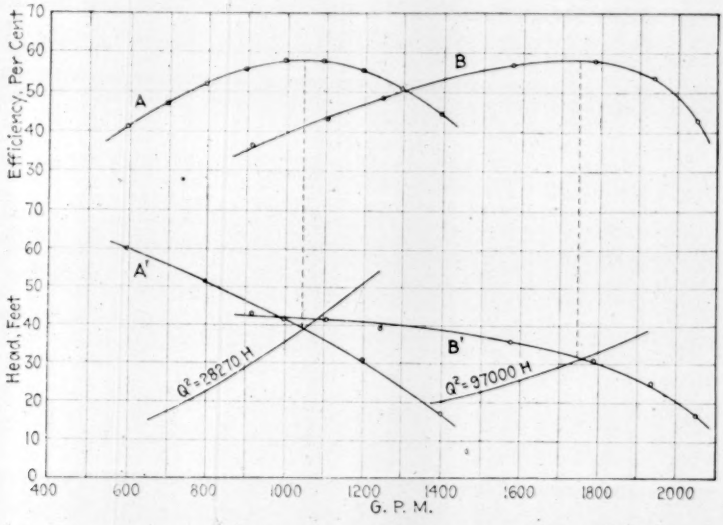

FIG. 24 CHANGES DUE TO DIFFERENCE IN IMPELLER DESIGN

series of pumps was designed. A typical section is shown in Fig. 22. An effort was made to mold the interior of the pump casing so that the water would change velocity and direction smoothly and with a minimum of eddy losses. The best position, however, of the cut-off is an uncertain matter. The intention is so to locate it that throughout the range of change in direction of water discharged from the impeller, there will be no eddy losses occasioned by the direction of stream lines crossing the cut-off. In the dotted lines, the general form of the volute suction inlet is shown. The initial advantage of this helicoidal type of runner is well set forth in the end view. The edges of the blades can be spaced and machined accurately and water can be taken into the impeller from the hub to the rim.

42 In Fig. 23 the elevation is partly shown in section. The feature strongly emphasized in the design of this pump is the ability to withdraw the impeller and shaft endwise from the casing simply by removing one bearing head. As compared with the split casing type of pump, this construction permits the suction and discharge to be located at any direction required by circumstances. Also, it frequently happens, as in the basement of a power plant or in cramped quarters on shipboard, that crane service is not available. This would make it difficult to remove the top half of a casing, while two men can handle the shaft and impeller even of a 30-in. pump of this type.

43 Of details of construction as shown in Fig. 23, the throat

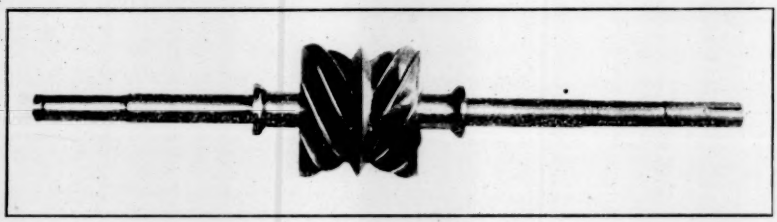

FIG. 25 TYPICAL NOTCHED IMPELLER

rings are of bronze, the sleeves on the shaft are locked right and left to hold the bronze impeller in any desired position and to prevent rusting of the shaft by the fluid pumped. The water discharged from the pump is used to seal the glands and prevent air drawing into the suction passage. The bearings are ring oiling with split babbitt-lined bearing shells. The design of the impeller permits balancing which is practically perfect, both at rest and running. This balance is not destroyed by the stresses set up by rapid rotation. The result has been that in the pumps so far constructed, the bearings ran with even less tremor or vibration than the bearings of the turbine used to drive them.

44 Something of the changes in pressure capacity curves which may be secured by suitable impeller design is shown by Fig. 24. The curves AA' and BB' result from a rim speed due to 3000 r.p.m. for a 6 in. diameter and a difference in helix angle and radial depth of blade. Theoretically, the parabola $Q^2 = KH$, may lie anywhere in the quadrant.

45 With the ordinary centrifugal impeller, change of capacity at a given head and speed is obtained by width of discharge

opening; in the helicoidal, the helix angle may be changed, greater capacity resulting from a greater angle. Change of head at a given speed and capacity may be obtained from the centrifugal impeller by change of blade angle at discharge or of diameter; but in the helicoidal, by notching as illustrated by Fig. 25, and if the relations of helix angle and depth of notch are suitable, the efficiency is increased by notching. Leakage back into suction is reduced by the screw conveyor effect of a short cylindrical portion at either side revolving within the throat

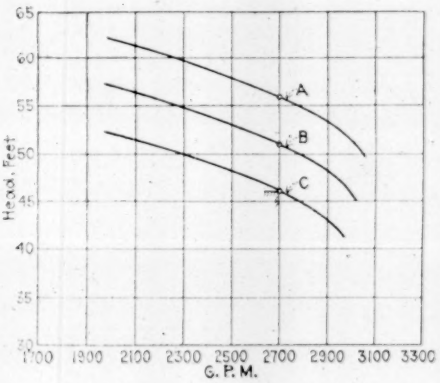

FIG. 26 CHANGES IN HEAD AT CONSTANT SPEED AND EFFICIENCY DUE TO NOTCHING IMPELLER

rings. The curves *A*, *B* and *C* of Fig. 26 show the result of beginning with the full helicoid and notching successively to obtain the rating of 2700 gal. per min. at 46 ft. head and the constant speed of 3000 r.p.m. The maximum efficiency was obtained at rated capacity throughout.

46 With the same axial inlet velocity assumed, the helicoidal impeller lies within the eye or inlet opening of the centrifugal. Then by notching deep with small helix angles, relatively low head and large volume may be obtained with good speed and efficiency. The general appearance of one form of such a pump is very well presented by Fig. 27, for which the impeller shown in Fig. 25 was constructed.

47 The derivation of the parabola showing the relation of head and capacity at maximum efficiency has already been explained. Reference has also been made to the fact that in surface condenser work where suction and discharge connections

are submerged, the total head may be entirely due to friction in tubes and piping. The general equation for friction head is $F = f\gamma s \frac{v^2}{2g}$, where $\frac{v^2}{2g}$ represents a *height;* as s is an area, $s.\frac{v^2}{2g}$ is a *volume;* and $\gamma s.\frac{v^2}{2g}$ is a *weight.* Then f is a numerical coefficient expressing the ratio of the force F required to overcome the friction of the weight on the surface. With area in sq. ft. and weight in lb., F is naturally expressed in lb.; but 1 lb. pressure per sq. in. is produced by a height of 2.31 ft. of water, and f may be replaced by f' so that the frictional resistance is finally

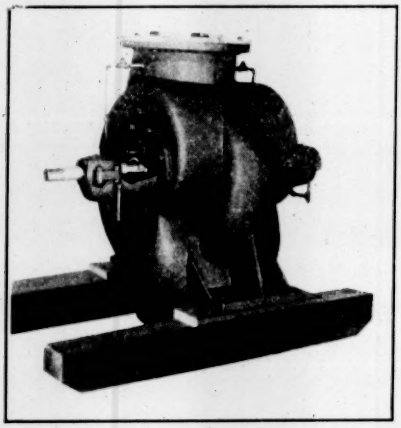

FIG. 27 TWENTY-INCH CIRCULATING PUMP

expressed as a total head in ft. Then solving $v^2 = \frac{2gF'}{f\gamma s} = K F'$, which is also the equation of a parabola. Now the capacity Q in gal. per min. is proportional to velocity v in ft. per sec. which produces the friction head. Hence if the parabola of head and capacity has a point in common with the friction parabola, they must coincide throughout as they have a common origin at zero head and capacity.

48 Assume that a maximum capacity of 11,000 gal. per min. is required at a total friction head of 20 ft. From this $K = \frac{11{,}000^2}{20} = 605 \times 10^6$, for slide rule calculation; and by assuming values of Q in $H = \frac{Q^2}{K}$ the head capacity parabola of maximum

efficiency in Fig. 28 may be determined. This may be associated with the 20-in. pump of Fig. 27 running at 1640 r.p.m., and such a pump properly fitted to conditions will operate at maximum efficiency regardless of actual speed and capacity.

49 But suppose the total friction head were found to be 15 ft. instead of 20 ft. Then the unit designed for 20 ft. head would deliver the required 11,000 gal. per min. at a speed of 1540 r.p.m. and an efficiency of 62 per cent instead of the maxi-

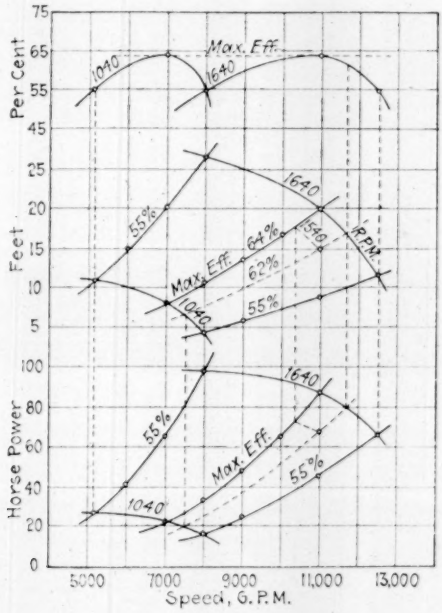

FIG. 28 PERFORMANCE CURVES FOR A CENTRIFUGAL PUMP AT MAXIMUM EFFICIENCY WITH VARIABLE SPEED

mum of 64 assumed. The dotted lines of Fig. 28 indicate the relative changes. Observe at this time that along these parabolas, whether of maximum efficiency or otherwise, the capacity varies with the speed, the head with the square and the power with the cube of the speed. And also that, if the submerged suction and discharge water levels are not at the same elevation, the parabolas of friction and of total head not having the same origin could not be made to coincide throughout.

50 In barometric condenser service and in irrigation and drainage work the friction head is of less importance. More

attention must be paid to large variation in static head. And for such work the rapidly varying head capacity curve characteristic of the high speed pump marks that type especially useful.

The data for the 8-in. and the 30-in. pump curves, Figs. 13 and 19, were obtained from tests conducted by Mr. Geo.A,Orrok, mechanical engineer for the New York Edison Company and his assistant engineers. The plan of testing the 30-in. pump was also approved by Prof. W. T. Magruder of the Ohio State University, and tests were witnessed by him. Duplicate tests were afterwards witnessed by Capt. F. H. Bailey, U. S. N.

TESTS UPON THE TRANSMISSION OF HEAT IN VACUUM EVAPORATORS

By E. W. Kerr

ABSTRACT OF PAPER

This paper deals with a series of tests made at the laboratories of the Louisiana State University to determine the effect of hydrostatic head, density of heating steam, incondensable gases in the heating steam, density of the liquid being boiled, design of the heating compartment, etc., upon the transmission of heat in vacuum evaporators of the kind used in the sugar and other industries.

The tests were made upon an apparatus designed especially for the experiments. Five different designs of heating compartments varying as regards tube dimensions, tube arrangement, circulation of juice, steam distribution and manner of removing incondensable gases were tested.

Some of the fundamental principles concerning the design and operation of multiple evaporators are discussed in the paper. It includes also various curves and tabulated data giving the results of the tests.

TESTS UPON THE TRANSMISSION OF HEAT IN VACUUM EVAPORATORS

By E. W. Kerr, Baton Rouge, La.

Member of the Society

The object of the tests described in this paper was to secure experimental data regarding the effect of the different factors which influence the transmission of heat in vacuum evaporators.

2 After making a number of tests on multiple evaporators in sugar factories it was concluded that a special experimental plant would have to be constructed for testing purposes, in order to control the conditions of operation. Accordingly, an apparatus was designed and built and, although considerably smaller than the average commercial machine, was large enough to give results comparable with what might be obtained from units of commercial size. This apparatus was erected and operated in the mechanical laboratory of the Louisiana State University and Audubon Sugar School.

3 The transmission of heat through the tubes of evaporators is affected by the following factors:

a Velocity of juice circulation
b Distribution and velocity of the heating steam
c Presence of incondensable gases in the steam compartment
d Hydrostatic head of the boiling liquid upon the heating surface
e Presence of condensed steam upon the heating surface
f Density of the liquid being concentrated
g Cleanliness of the heating tubes

4 *Velocity of Juice Circulation.* The circulation of liquid in most evaporators is not positive, but depends upon convection currents much the same as in steam boilers, steam tube evapora-

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tors being analogous to fire tube, and liquid tube evaporators to water tube boilers in this respect. In view of this fact it is evident that the advantages of high velocity of circulation are not obtained with evaporators to the extent possible with surface condensers, where the circulation of the cooling water is positively controlled by a pump or other means.

5 With submerged tube evaporators, the velocity of circulation can, however, be considerably increased by careful attention to details such as proportions of tubes and the use of circulation tubes, or downtakes. Fig. 4, one of the experimental evaporators used in the experiments, shows such a circulation tube at the center. The boiling liquid ascends through the small tubes and descends through the downtake tube, due to the fact that there is a greater amount of heat transmission per unit of carrying capacity through the small tubes than in the larger circulation tube.

6 *Distribution and Velocity of the Heating Steam.* Of the devices used in commercial evaporators for effecting even steam distribution to the heating surface may be mentioned the annular steam belt surrounding the tubular cluster, with slots through which steam passes radially inward. This has been used very generally with vertical submerged tube evaporators.

7 The vertical submerged tube (standard) evaporator with steam supplied through an annular belt offers little opportunity for securing high steam velocity; in fact, the average steam velocity is very low in this type. One of the evaporators experimented with, shown in Fig. 7, is opposite in principle to the belt type. It was designed to secure high velocity at the expense, however, of considerable friction, instead of low velocity and little friction as in the belt type. The steam tube evaporator gives opportunity for controlling steam velocities within certain limits by proportioning the tubes, a long tube of small diameter giving a greater velocity than a shorter tube of larger diameter.

8 *Presence of Incondensable Gases in the Steam Compartment.* The presence of air or other incondensable gases not only reduces the coefficient of heat transmission because of their resistance to heat conduction, but according to Dalton's law of mixed vapors, produces a temperature in the steam compartment lower than that corresponding to the vacuum shown by the gage and so decreases the temperature fall.

9 Most evaporators are provided with means for removing the

incondensable gases, though in many of them this is done more or less ineffectively because under the agitated conditions incident to the rapid inrush of steam there can be but little segregation. The removal of air must therefore be at the expense of much steam along with it. Incondensable gases are likely to collect in dead spaces which makes the matter of steam distribution of added importance.

10 Fig. 7 illustrates a special device applied to vertical liquid tube evaporators, one object of which is to effect a more efficient removal of the incondensable gases. As the steam passes toward the center of the steam compartment it condenses and gradually becomes richer in air, thus affording a means of separating air from steam, the former being drawn off from the center.

11 Fig. 6 is a steam tube evaporator especially designed to remove incondensable gases effectively. In this case the gases are removed from each steam tube, which is closed at the top, by means of a $\frac{1}{8}$-in. pipe. One type of film steam tube evaporator with horizontal tubes vents the gases directly to the vapor space of the boiling liquid through very small holes in the closed end of the tubes.

12 *Hydrostatic Head of the Liquid upon the Heating Surface.* Hydrostatic head of juice, that is, deep submergence of the heating surface, reduces heat transmission and capacity by decreasing the temperature fall for a given vapor pressure difference. The added pressure due to hydrostatic head increases the average temperature of the boiling liquid, and this in turn decreases the temperature fall. For example, assume the steam pressure in the calandria to be 20 lb. absolute and the pressure of the vapor above the boiling liquid 12 lb. absolute, corresponding to temperatures of 228 deg. and 202 deg. respectively. If the evaporator were of the film type, the actual temperature fall would be 26 deg. If it were of the submerged tube type like that of Fig. 5, with the juice level say 48 in. above the lower tube plate then the total pressure of boiling would be equal to the vapor pressure plus the average static liquid pressure. The latter is equal to $\frac{24}{12} \times 0.43 = 0.86$ and the total pressure to $12 + 0.86 = 12.86$ lb. per sq. in., corresponding to a temperature of 205 deg. in which case the temperature fall would be 228 deg. — 205 deg. = 23 deg. instead of 26 deg., as in a film evaporator.

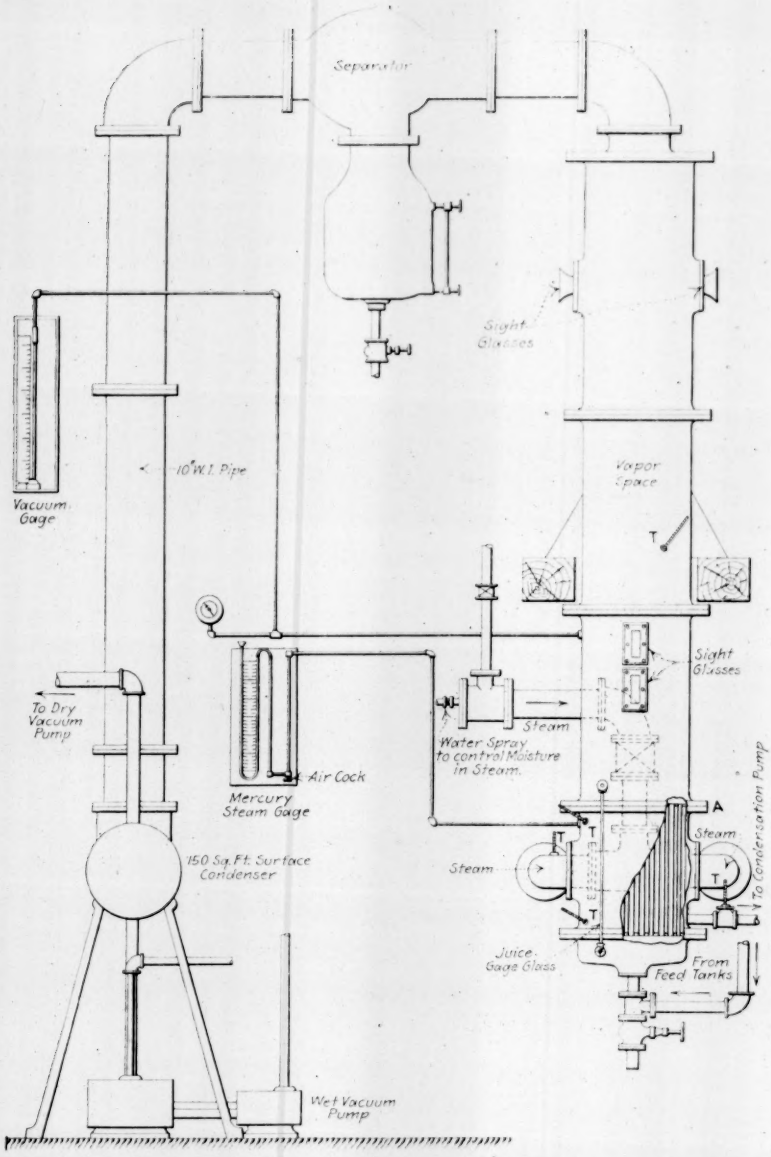


FIG. 1 GENERAL ARRANGEMENT OF EXPERIMENTAL VACUUM EVAPORATOR

A similar calculation will show that the lower the absolute pressure of boiling, the greater the loss in temperature fall with a given depth of submergence, and this means that the loss will be greatest in the last body of a multiple evaporator.

13 *Presence of Condensed Steam upon the Heating Surface.* As with all steam heating apparatus it is important that all condensed steam be removed as it is formed. If allowed to collect

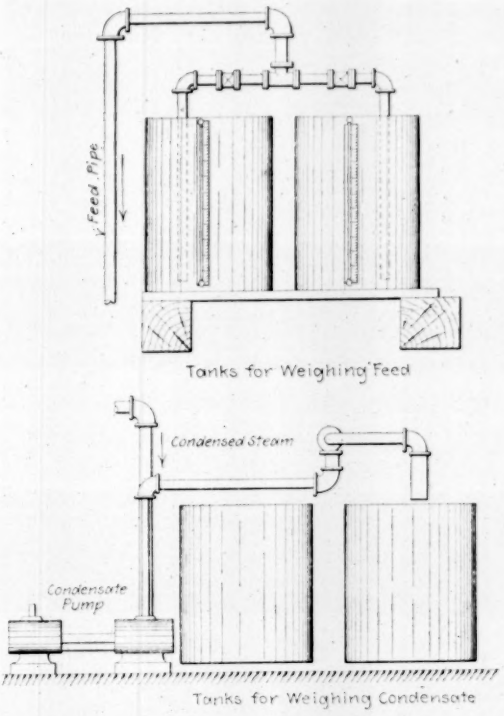
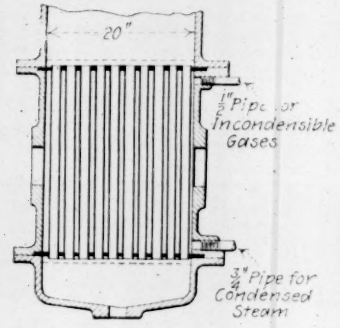


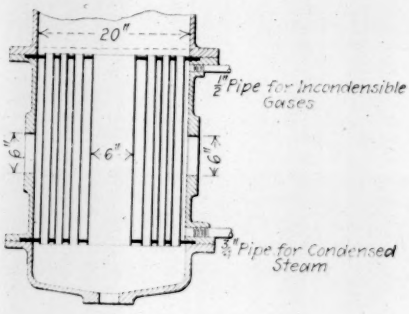
FIG. 2 WEIGHING TANKS FOR EXPERIMENTAL APPARATUS

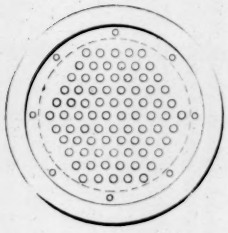
in the heating compartment the heating surface thus submerged is made less effective since the transmission from liquid to liquid is less than from steam to liquid. In evaporators with long vertical tubes the condensed steam must run down the entire length of the tubes, and this doubtless reduces the heat transmission. With horizontal steam tubes, the water is swept out of the tubes by the current of steam.

14 *Density of the Liquid being Concentrated.* The density of the liquid being concentrated affects heat transmission in that

the temperature of boiling is increased above that corresponding to the pressure and this results in decreased temperature fall. The coefficient of heat transmission may also be decreased in some cases, though this is a matter of some doubt. The effect of density in this respect is naturally greatest in the later bodies







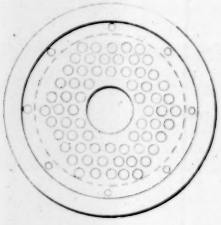


FIG. 3 FIG. 4

FIG. 3 80 COPPER TUBES $1\frac{3}{8}$ IN. BY 24 IN. BY 18 BIRMINGHAM WIRE GAGE
FIG. 4 66 COPPER TUBES $1\frac{3}{8}$ IN. BY 24 IN. BY 18 BIRMINGHAM WIRE GAGE. 6 IN. DOWNTAKE. TUBE PLATES $\frac{3}{16}$ IN. COPPER

where the absolute pressure is least and where the density is greatest.

15 *Cleanliness of the Heating Tubes.* The effect of the fouling of heating tubes upon heat transmission is too well known to be remarked. The collection of solid matter upon the heating surface depends mainly upon the nature of the liquid and the velocity of circulation. With sugar juices it is generally necessary to stop the evaporation every one or two weeks to clean the tubes. This may be done by boiling out with caustic soda followed by hydrochloric acid. If exhaust steam is used for heating, there is likely to be some fouling due to the lubricating oil contained, especially if cheap oils, which saponify under the action of heat, are used.

DESCRIPTION OF APPARATUS

16 As it is very difficult to control conditions in the different bodies of a multiple evaporator, it was decided to use an evaporator with a single body so constructed that the conditions existing in any of the bodies of a multiple evaporator could be reproduced. With this in view the apparatus was so designed that it was possible to change the type of heating compartment and to control at will the pressure of the heating steam, the vacuum under which boiling takes place, the quality of the steam, the amount of air in the heating steam, and the density of the liquid.

17 Figs. 1 and 2 show the general arrangement of the plant used in all of the tests. The surface condenser contains 150 sq. ft. of cooling surface, whereas the heating surface in the evaporator itself varied from 50 sq. ft. in the smallest calandria tested up to 86 sq. ft. in the largest. It will be noted that the body of the evaporator is supported above the calandria so that the calandria can be removed and another one bolted on conveniently. Fig. 1 shows the calandria of Fig. 3 bolted to the body at *A*. In changing from one calandria to another this joint had to be broken.

18 Figs. 3 to 7 inclusive show the different types of calandria tested. The calandria shown in Fig. 4 is like that of Fig. 3 except that there is a 6-in. circulation tube or downtake at the center. Fig. 5 shows another calandria with tubes 2 in. in diameter and 48 in. long without a downtake.

19 Fig. 6 shows an evaporator of the steam tube type with special arrangements for removing the incondensable gases. There are two thick tube plates, one above the other, both of cast iron. The heating tubes which are 2 in. in diameter and 54 in. long are expanded into the upper tube plate. These tubes are closed at the top and are open at the bottom. Into the bottom tube plate $\frac{1}{8}$-in. tubes are screwed. These small tubes are open at both ends and are placed inside of the heating tubes, reaching nearly to the top of the latter. The space below the lower tube plate is connected to the vapor space of the succeeding body in an evaporator of commercial size. Steam passes up into the heating tubes in the annular spaces surrounding the gas tubes, the incondensable gases being driven towards the top from whence they are removed by the small gas tubes, each heating tube having

its own individual incondensable gas remover. The lower tube plate is made saucer shape so that the condensed steam drains to the center from where it is removed in the usual manner.

20 Fig. 7 shows a calandria similar to Figs. 3 and 4 with tubes 1¾ in. in diameter and 24 in. long. It differs from them,

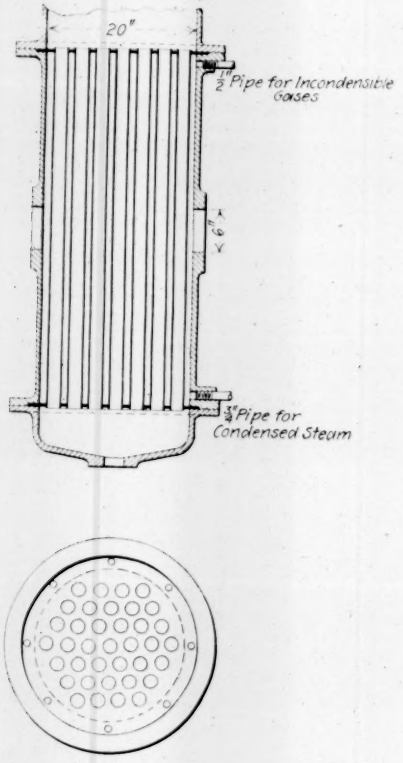


FIG. 5 37 COPPER TUBES 2 IN. BY 48 IN. BY 18 BIRMINGHAM WIRE GAGE TUBE PLATES 3/16 IN. COPPER

however, in the manner of distributing the steam to the heating tubes, and in the manner of removing the incondensable gases, the steam being supplied through four openings, one above the other, thus giving better distribution vertically. Between the two tube plates a vertical baffle plate is placed so as to guide the steam along a circuitous path to the 3-in. downtake at the center. This baffle plate is so placed that the passage for the steam is gradually reduced in cross-section in order to keep the steam velocity

as high as possible, overcoming to an extent, the tendency to decrease the velocity due to condensation. The incondensable gases are drawn off by means of a small perforated pipe through the

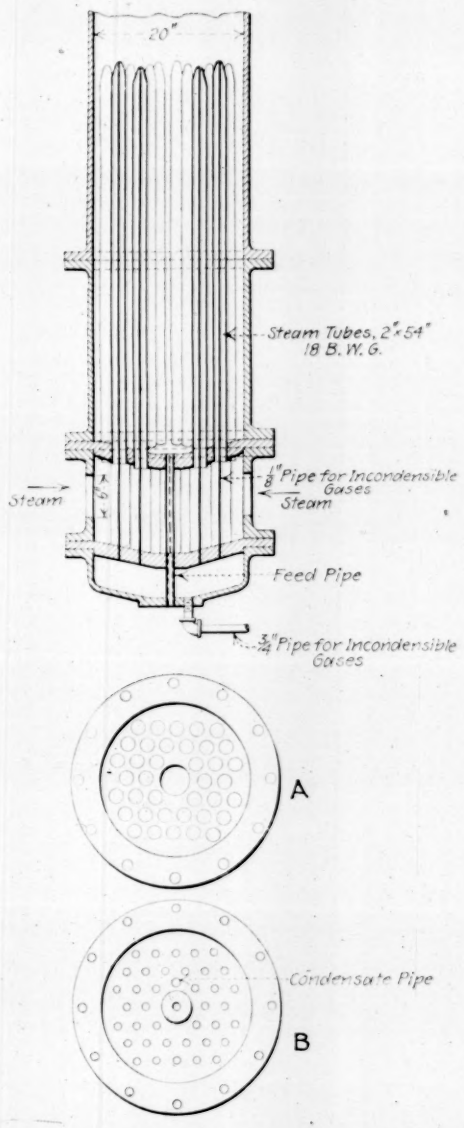


FIG. 6 38 COPPER HEATING TUBES 2 IN. BY 54 IN. BY 18 BIRMINGHAM WIRE GAGE. TOTAL HEATING SURFACE 86.21 SQ. FT.

top tube plate and reaching nearly to the bottom tube plate. The object of this design is to obtain high steam velocity among the heating tubes and effective separation of incondensable gases from steam.

21 The tanks for weighing the condensed steam, or "con-

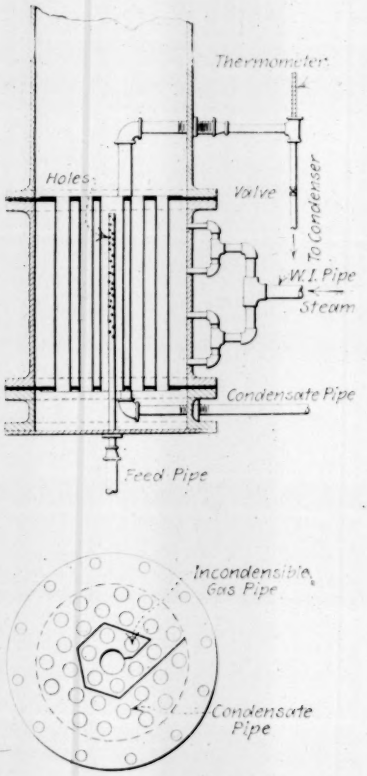


FIG. 7 .30 COPPER HEATING TUBES $1\frac{1}{4}$ IN. BY 24 IN. BY 18 BIRMINGHAM WIRE GAGE. TOTAL HEATING SURFACE 29 SQ. FT.

densate," were used in only a few of the tests in which it was attempted to determine the radiation loss from the surface of the evaporator. The steam used in the evaporator was taken from the power house boiler at a pressure of about 80 lb. gage, the desired pressure in the tube belt being obtained by throttling at the valve.

22 The steam pipe in all of the calandrias except that of Fig. 7 was 6 in. in diameter at the entrance to the tube section and

made to enter at two points. In this way the steam was well distributed. The steam being expanded in some cases from 80 lb. down to less than atmospheric pressure, there was considerable superheat. In order to get steam of varying quality provision was made for spraying water into the steam between the throttling valve and the calandria.

23 In commercial evaporators it is customary to remove the concentrated liquor from the last body by means of a pump. In the experimental outfit a slightly different arrangement was used. In starting a test, sugar juice of the desired density and quantity was placed in the evaporator and as the evaporation progressed water was supplied in such quantities as would keep a constant level in the juice compartment, as shown by the juice gage glass. In this manner conditions sufficiently near those in actual evaporators were obtained with considerably less complexity of apparatus. After being weighed in the calibrated barrels, the water was fed by gravity aided by the vacuum in which the boiling was usually carried on, a valve in the feed pipe being used to regulate it.

24 That part of the evaporator above the top tube plate, which will be designated as vapor space, was 10 ft. high in all of the calandrias tested except that of Fig. 6 in which it was about 8 ft. This liberal height was provided in order to prevent, as far as possible, the carrying over of liquid in the vapors leaving the boiling surface. The separator shown was originally designed to be used as an oil separator, but was here used to catch any liquid entering the vapor pipe.

25 Thermometers were placed as shown in Fig. 1 to measure the temperatures in the steam pipe, the vapor space, the bottom of the steam compartment, the top of the steam compartment, the entering juice or water, the condensed steam and the room. This applies in its entirety to the apparatus when used with the tube sections shown in Figs. 3, 4 and 5. The thermometers were placed somewhat differently for the other tube sections. The mercury manometer connected to the steam compartment could be used with pressure either above or below the atmosphere and was arranged so that the water could be kept out of the mercury column when operating with pressure. With low vacua the wet vacuum pump was sufficient, but with higher vacua, that is, 24 in. or more, it was necessary to use the dry vacuum pump. A ma-

jority of the tests was made with water as the liquid to be boiled, the balance being made with sugar solutions produced by mixing white sugar with water in such proportions as would give desired densities. Liquor made in this manner was clean and pure and gave practically no fouling of the heating tubes. All tests were made with practically clean heating tubes.

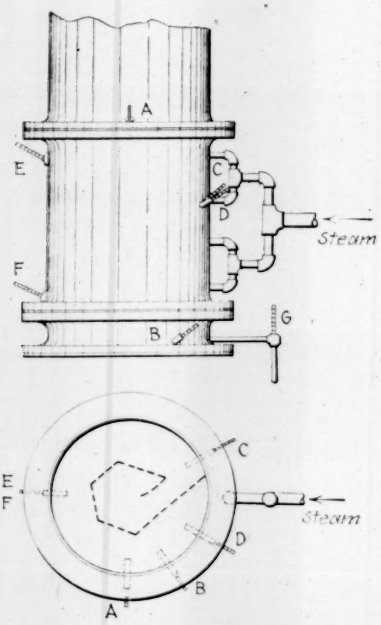


FIG. 8.—DIAGRAM SHOWING LOCATION OF THERMOMETERS CALANDRIA E

METHOD OF MAKING A TEST

26 In starting a test the wet vacuum pump was first started and a charge of juice drawn in. The condensate pump was then started and steam turned on to warm the apparatus. After the desired conditions as regards steam pressure, vacuum, height of boiling, etc., were obtained the apparatus was operated for some time before the test was started. The duration of the tests varied from 20 to 60 minutes, depending upon conditions. Each test started with the juice level indicated by a string around the juice gage glass and the test ended with the same level. If the boiling was not very rapid and violent the level, as shown by the gage

glass, could be read with sufficient accuracy to enable short tests to be run. When there was more rapid boiling tests of longer duration became necessary.

27 Readings of the instruments were taken every 5 minutes throughout the tests.

EFFECT OF HYDROSTATIC HEAD

28 Tests were made on calandrias A, C, D and E in order to secure data on the effect of hydrostatic head. Four series of

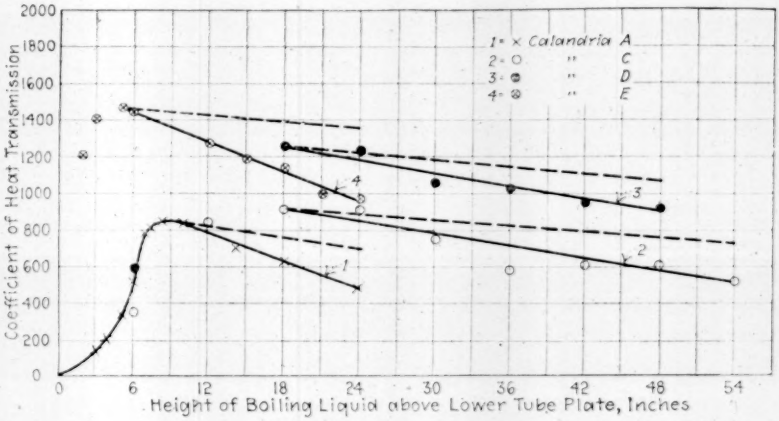


FIG. 9 CURVES SHOWING EFFECT OF HYDROSTATIC HEAD UPON HEAT TRANSMISSION. FULL LINES PLOTTED FROM TEST; BROKEN LINES DRAWN ACCORDING TO CALCULATIONS BASED UPON THEORETICAL LOSS OF TEMPERATURE FALL DUE TO HYDROSTATIC HEAD

tests were run under practically the same conditions of steam and boiling pressures. The height of boiling was varied in each series and other conditions kept constant, or as nearly so as possible. Water was used for boiling in these tests. The results are given in Table 2 of the appendix and are plotted in Fig. 9 herewith.

29 The general procedure in each of these series was to start with very low heads, a test being run for each head and each succeeding test being made with increased head until the desired maximum was reached. At very low heads only the lower portions of the tubes were kept wet by the boiling water, hence more or less of the heating surface was ineffective, the maximum evaporation being obtained with the head at which the ebullition

was sufficient to project the boiling water just to the tops of the tubes. Further increase of head showed a decrease in the rate of evaporation and in the coefficient of heat transmission.

30 The left-hand end of each of the curves plotted in Fig. 9 represents as nearly as can be determined this maximum evaporation. Points obtained with lower heads than this are plotted without drawing curves through them with the exception of Curve 1. The broken lines represent the theoretical decrease in heat transmission due to increase in head, determined as described in Par. 12. It will be noted that the actual curves drop below the theoretical. Just why this is so cannot be determined with certainty, although it may be due to a decrease in the velocity of liquid circulation as the head is increased.

31 Curves 1 and 4, Fig. 9, were both obtained from tests on calandrias with tubes 24 in. long. It will be noted that these two curves are practically parallel. Curve 2 was plotted from tests upon a liquid tube apparatus with tubes 48 in. long, while Curve 3 was plotted from tests upon a calandria having steam tubes 54 in. long. It will be noted that these two curves are also practically parallel to each other. It would thus seem that with tubes having the same length the rate of decrease in evaporation due to hydrostatic head is practically constant. It will be noted further that the curves for the 24-in. tubes are much steeper than those for the longer tubes, showing that the effect of hydrostatic head is greater with short tubes than with longer ones. This is also difficult to explain, although circulation doubtless has something to do with it.

EFFECT OF TEMPERATURE LEVEL

32 The total temperature fall in a multiple effect may be increased by increasing the temperature, or what is the same thing, increasing the pressure of the steam supplied to the first body, or by decreasing the pressure and temperature in the vapor space of the last body. It is evident that increasing the temperature fall, and therefore the capacity, by the first method results in increasing the average temperature in the heating compartments and that the second method results in decreasing it.

33 The relative advantages of high steam pressure in the first body as compared with high vacuum in the last body in obtaining increased heat transmission has been a matter of some question by many. Then, too, the reason for the inequality of temperature

fall in the different bodies of a multiple effect, the greatest fall being always in the last body where the temperature level is lowest, is not definitely known, although it has been thought that the lowest steam density in the last body might be partly responsible.

34 In order to get data that would aid in settling these points a large number of tests were made which are given in Table 3. These tests consist of five series B-1 to B-5 inclusive, each series being made upon a different calandria, but in which the temperature level was varied in practically the same manner, all other conditions in each series being maintained as nearly constant as was possible. The limits of temperature in calandria, although varying somewhat in the different series, approximate those with which multiple effects are operated in practice.

35 The relations between temperature of steam in calandria and coefficient of heat transmission are shown graphically in the curves of Fig. 10. The downward trend of the curves in the low temperature regions gives unmistakable evidence that the lower the temperature level the lower the coefficient of heat transmission, other things being equal. It is most likely that this is due to the lower density of the steam average. Fig. 11 gives an average of the five curves of Fig. 10, but with steam densities instead of temperatures as abscissas. Some of the falling off in capacity at the low pressures may be due to air in the steam. There would naturally be more leakage as the vacuum increased as well as a greater proportion of air to steam. All of the tests were made, however, with tight joints and with the observed temperatures of steam in calandria indicating no partial air pressure. The variation in height of the curves in Fig. 10 is due to various reasons which will be discussed later.

EFFECT OF INCONDENSABLE GASES ON HEAT TRANSMISSION

36 For the purpose of studying the influence of incondensable gases a series of tests was made on calandria A, Fig. 3. The amount of air present in the heating steam was regulated by varying the speed of the condensate pump and by admitting air into the steam compartment through a pet cock shown in Fig. 1. The quantity of air present was determined by the temperature method. Through actual observation of the temperatures in the steam compartment the partial steam pressure was obtained and this subtracted from the gage pressure gave the partial pressure

due to air. The location of the thermometers used for measuring the temperatures in the steam compartment are shown in Fig. 1, two thermometers being used, one near the bottom and one near the top, both of which were about 45 deg. from the left steam pipe entrance. Thermometer wells containing mercury and extending about 2 in. into the steam space were used. The average reading of these two thermometers was assumed to be the temperature in the steam compartment.

37 The results of these tests are given in Table 4 of the ap-

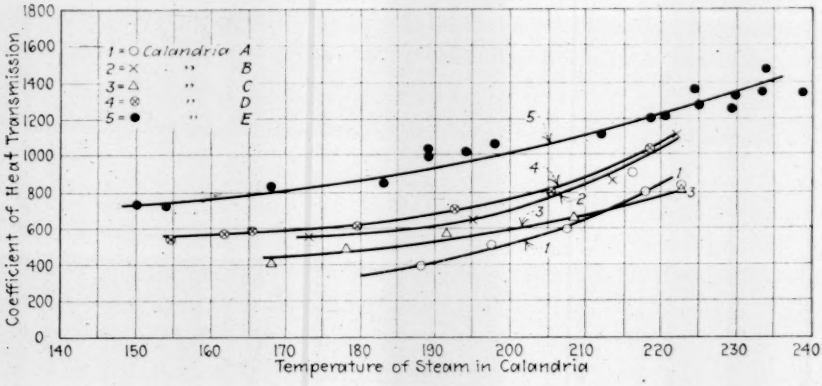


FIG. 10 CURVES SHOWING EFFECT OF TEMPERATURE LEVEL UPON HEAT TRANSMISSION IN FIVE DIFFERENT CALANDRIAS

pendix. Fig. 12 shows the results graphically, $\frac{P_s}{P_t}$ being the ratio of the partial steam pressure to the observed gage pressure. In the equation for the curve

$$U = \left(\frac{P_s}{P_t}\right)^n$$

the exponent n has a value of nearly 3, showing only a slightly greater decrease of the coefficient due to air than in the tests by George A. Orrok[1] on the effect of air in surface condensation in which the value of 2 was given to n. The absolute pressure in the evaporator tests was about 16.5 in. of mercury, whereas in the Orrok tests it was only about 2 in. These evaporator tests differed also from the condenser tests referred to as regards hy-

[1] Air in Surface Condensation, Geo. A. Orrok, Trans. Am. Soc. M. E., vol. 34, p. 713.

drostatic head and the range of the values of the coefficient of heat transmission. In the Orrok tests the maximum coefficient was only slightly above 300 while in the evaporator tests it was above 500. This latter was probably due to the greater steam density in the evaporator tests.

38 Attention has already been called to the fact that the evaporators shown in Figs. 6 and 7 were designed for the especial purpose of overcoming the bad effects of incondensable gases. While it was practically impossible to determine the amount of air present, for the reason that the temperature existing adjacent to the heating surface could not be measured, the results obtained

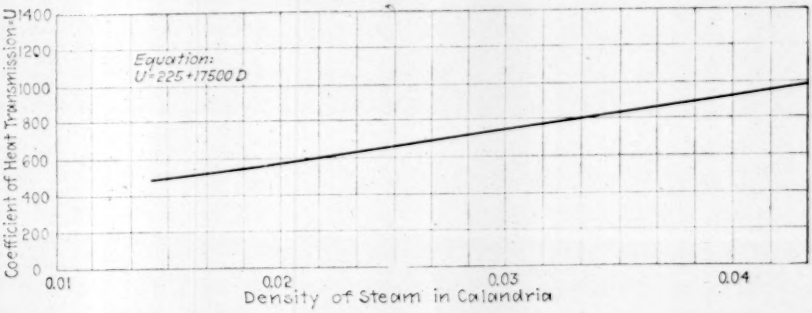


FIG. 11 CURVE SHOWING VARIATION IN HEAT TRANSMISSION WITH VARYING DENSITIES OF HEATING STEAM

with these two arrangements, show considerable improvement in heat transmission over those obtained from the others. This is shown roughly in Fig. 9 by the relative height of the Curves 1 and 4, also Curves 2 and 3. The increase is doubtless due mainly to the more effective separation of air from the steam and to its prompt removal from the heating compartment. The curves shown in Figs. 10 and 15 also show the increased heat transmission in these two calandrias.

39 It will be noticed that the steam used in the tests plotted in Fig. 12 was superheated and although the temperatures in the calandria were taken where steam and condensation were in more or less intimate contact the temperatures found may have been slightly above what would have existed if saturated steam had been used. This probably accounts for the fact that the upper end of the curve shows no air, whereas it is reasonable to suppose there was a small amount present. In view of this fact

the position of the curve may be slightly inaccurate and its value comes mainly from its direction. In several tests on calandria D and calandria E large amounts of air were admitted through the pet cock used for admitting air in the series plotted in Fig. 12. In fact, judging from the amount the pet cock was open, more air was admitted than in any of the tests plotted in the series.

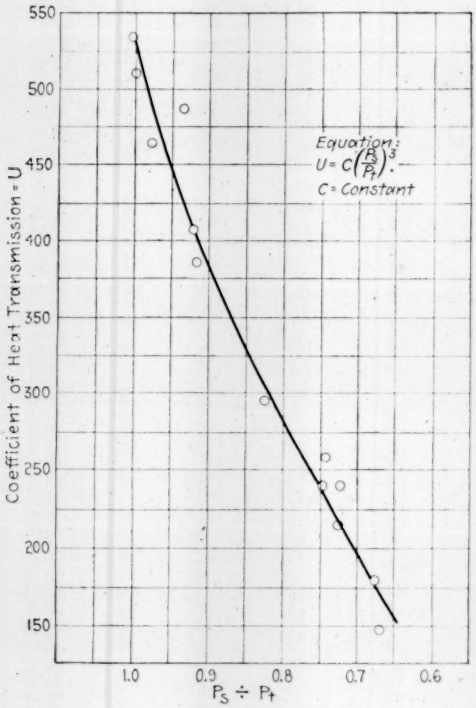


FIG. 12 CURVE SHOWING EFFECT OF AIR UPON HEAT TRANSMISSION

It was found, however, that such admission of air had little effect in reducing the coefficient of heat transmission.

40 The thermometers inserted in the steam space of calandrias A, B, and C, where temperatures were taken both at the top and bottom, seldom gave equal readings except when there was no air present. This condition occurred only when the observed temperature was equal to or greater than the saturation temperature corresponding to the observed pressure. In the tests on calandria A, the greater temperature was in practically all cases reg-

istered by the thermometer at the top, whereas with calandrias B and C the reverse was the case.

41 In the tests on calandria E. thermometer wells were in-

TABLE 1 VARIED TEMPERATURE CONDITIONS IN CALANDRIA DUE TO PRESENCE OF AIR AND SUPERHEAT

Test Number	Absolute Pressure in Calandria, In. Mercury	Saturation Temperature due to Pressure in Calandria	Temperature, Deg. Fahr., Steam in Calandria				
			at D*	at F*	at E*	at C*	at H*
60	7.6	150.1	150.0	150.4	151.8	151.5	147.3
61	7.99	152.2	151.4	156.4	153.9	154.5	148.8
62	11.12	166.0	165.9	172.3	167.6	170.8	160.3
70	34.75	219.5	221.3	220.7	221.9	219.4	215.2
73	39.84	226.6	229.0	229.6	230.3	230.0	222.6
74	40.55	227.6	230.17	230.8	228.7	231.4	223.7
75	32.10	230.8	234.4	233.0	234.7	232.3	226.1
76	43.57	231.5	235.4	233.8	235.6	234.0	227.0

*See Figs. 7 and 8.

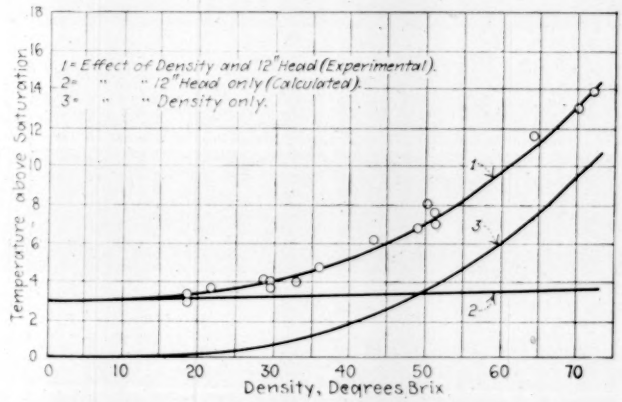


FIG. 13 CURVES SHOWING EFFECT OF DENSITY OF LIQUID UPON BOILING TEMPERATURE

serted at various parts of the calandria as shown in Figs. 7 and 8 for the purpose of observing steam temperatures. Table 1 shows some interesting data selected at random to give an idea of the unstable conditions due to the presence of air and superheat as shown by the variation of temperature at different points and times.

DENSITY OF THE LIQUID

42 The density of liquids may be determined by either of two hydrometers, viz., Brix or Beaumé. In these tests a Brix hydrometer was used. The following tabulation shows relations which may be of use in grasping the meaning of the data given in Table 5.

Brix	Beaumé	Specific Gravity
0	0	1.0
10	5.7	1.0401
20	11.3	1.0833
30	16.8	1.1296
40	22.3	1.1794
50	27.7	1.2327
60	33.0	1.2899
70	38.1	1.3509

The Brix spindle reads directly the per cent of solids in a solution.

43 Increasing the density of the liquid being boiled decreases the heat transmission by decreasing the temperature fall. The circulation is also affected by the density and this also affects heat transmission. The boiling temperature of solutions increases with the density and in order to get first-hand data on this subject, a series of tests was made upon calandria E, the density of the sugar juice being varied from 18 to 70 Brix and the height of the juice kept at 12 in. The results of these tests are shown in series D-1, Table 5. By subtracting the saturation temperature corresponding to the pressure on the boiling liquid from the observed temperature of the boiling liquid the increase in the boiling temperature due to the combined effect of hydrostatic head and juice density was obtained. Curve 1, Fig. 13, was obtained by plotting these against the corresponding densities. Curve 2 represents the calculated loss in temperature fall due to the 12 in. hydrostatic head for the conditions of the tests. By subtracting the ordinates of Curve 2 from corresponding ones of Curve 1, Curve 3, representing the loss due to density alone, was obtained. The equation of this curve is $y = CD^{3.1}$ in which C = a constant and D = density of liquid, Brix. These tests were very carefully made after many preliminary tests for the purpose of controlling conditions properly and it is believed that the true relation between density and decrease in temperature fall is shown in Curve 3.

44 Another set of tests was made to determine the actual variation in heat transmission as affected by density. The data from these tests is given in series D-2 of Table 5 and shown graphically in Fig. 14.

TYPE OF CALANDRIA

45 The five calandrias tested differ from each other in several fundamental features to which attention has already been called. Most important among these differences of design may be mentioned proportions of tubes, that is, ratio of length to diameter; the use or non-use of downtakes to aid the circulation of the juice and the methods employed for removing incondensable gases. What appears to be more or less conclusive data bearing upon the relative merits of these different designs were obtained in the tests. This is probably brought out best in the hydrostatic tests, Fig. 9, also in the temperature level tests, Fig. 10. Both of these sets of curves, however, show the effects of difference of design only in the rough for the reason that the conditions of operation were not the same in all of the tests. In the case of the temperature level tests there was considerable variation in the hydrostatic head. In order to make these tests more nearly comparable the curves of Fig. 10 are reproduced in Fig. 15

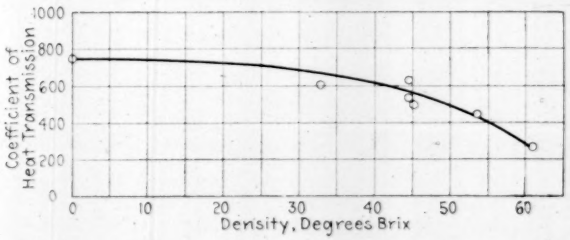


FIG. 14 CURVE SHOWING EFFECT OF DENSITY OF LIQUID UPON HEAT TRANSMISSION

with the necessary corrections for hydrostatic head based upon the curves of Fig. 9. The full line curves are taken without change from Fig. 10. The broken line curves are those corrected for hydrostatic head. The principal conditions during the tests are noted on the plat.

46 Curves 3 and 5 show something as to the effect of varying the proportions of the tubes, the former for tubes 1⅜ in. by 24

in. and the latter for tubes 2 in. by 48 in., some advantage being shown for the long tubes, at least for low temperature levels. The greater heat transmission in the long tubes is probably due to better circulation. The greater the ratio of the length of a tube to its cross-sectional or carrying area, the greater will be the heat transfer per unit of carrying area and this should increase the velocity of circulation.

47 A comparison of Curves 3 and 4 shows a decided advantage for the downtake calandria over that without a downtake. The tests on the downtake calandria were made with a hydrostatic head of 16 in., whereas those on the other were made with

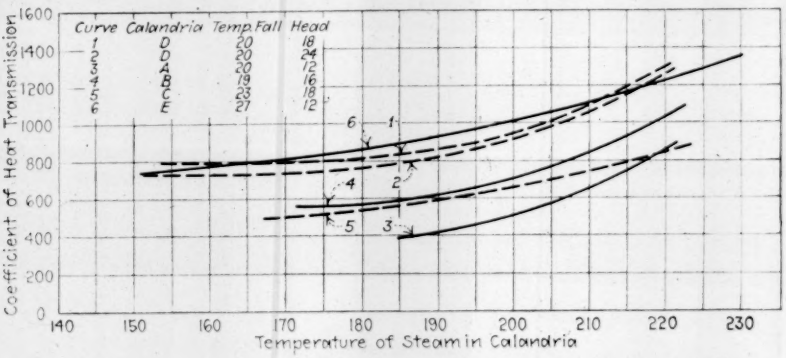


FIG. 15 CURVES COMPARING HEAT TRANSMISSION IN DIFFERENT CALANDRIAS TESTED. FULL LINE CURVES COPIED FROM FIG. 10; BROKEN LINE CURVES SAME AS IN FIG. 10 AFTER CORRECTION FOR HYDROSTATIC HEAD

a 12 in. head. Correction for this difference would slightly increase the advantage shown for the downtake calandria.

48 Curves 1 and 6 show the results of the tests on two types of evaporators designed especially for efficient removal of incondensable gases. Both show coefficients of heat transmission considerably in excess of those obtained with the standard types represented by Curves 3, 4 and 5. Curve 6 for calandria E, the baffle plate calandria with a head of 12 in., is only slightly higher than Curve 1 which represents D, the double tube calandria with an 18 in. head. These relations are also corroborated in a general way by the curves of Fig. 9. While the good results shown for these two calandrias may undoubtedly be attributed mainly to the more efficient separation of incondensable gases from steam and its removal, it is probable that the increased velocity of steam is also partly responsible.

CONCLUSIONS

49 The loss in heat transmission due to hydrostatic head is considerably in excess of the theoretical, and this excess is greater for short tubes than for long tubes. The loss with varying heads, other conditions being equal, varies according to a straight line formula.

50 Other conditions being equal, the lower the "temperature level" the smaller the coefficient of heat transmission, in other words, the lower the temperature and density of the heating steam the smaller the coefficient. The coefficient of heat transmission varies according to the equation

$$U = 2.25 + 17500\,D$$

where D = density of heating steam in pounds per cubic foot.

51 Air or other incondensable gases in the heating steam greatly reduce the heat transmission, even with relatively low vacua. The coefficient of heat transmission varies according to the equation

$$U = C\frac{P_s}{P_t}$$

in which C is a constant, P_s the partial pressure of the steam and P_t the total pressure. The presence of "air pockets" may be conveniently determined by means of thermometers in the steam compartment: good circulation and distribution of steam are important in preventing them.

52 Increasing the density of the boiling liquid causes a loss in heat transmission due to the decrease in temperature fall. As the density is increased the boiling temperature increases according to the equation $y = CD^{3.1}$, in which C = a constant and D = density in deg. Brix. The total loss due to the density of the boiling liquid seems to be in excess of that due to loss of temperature fall. This is probably due to lower velocity of circulation.

53 The great temperature fall required in the last body of a multiple evaporator is due to the combined influence of greater amounts of air, steam of lower density, liquid of higher density, also in many cases, more foul heating surfaces, than in preceding bodies.

54 The downtake or circulation tube was shown to increase heat transmission materially. Long tubes give better results as to heat transmission than short tubes due to better circulation.

55 The tests show that the double tube and the baffle plate calandrias gave greatly increased heat transmission as compared with the standard types tested, and indicate that attention to steam distribution and the removal of incondensable gases is very important.

56 In closing, the author desires to acknowledge the valuable assistance of Messrs. J. A. Gunther and A. J. Isacks, who did most of the testing and calculating. Thanks is due Mr. T. F. Sanborn of New York City who built and furnished the double tube calandria, Fig. 6, and Mr. A. L. Weber of New Orleans who built and furnished the baffle plate calandria, Fig. 7. Some 350 tests have been made on the apparatus described, though this paper includes only those fulfilling required conditions.

APPENDIX

57 In the tables which follow will be found average observed and calculated data taken during the tests. In calandrias A, B and C, temperatures in the steam spaces were obtained by means of thermometers placed both at the bottom and at the top, and the item "observed temperature" given in the tables is the average of the readings from these two temperatures.

58 The temperature fall was found by subtracting the observed temperature in the steam space from the saturation temperature corresponding to the absolute pressure of the heating steam when the observed temperature in the calandria was equal to or greater than the saturation temperature corresponding to the pressure. In case the observed temperature in the calandria was less than the saturation temperature the temperature fall was obtained by subtracting the observed temperature in the vapor space from the observed temperature in the calandria. It was thought best to use saturation temperature in the calandria where superheat was shown, as condensation can take place only at saturation temperatures.

59 The coefficient of heat transmission, that is, "B.t.u. transmitted per square foot of heating surface per degree of temperature fall per hour," was obtained by dividing the B.t.u. transmitted per square foot per hour by the temperature fall. Marks and Davis steam table was used in working up all the tests.

TABLE 2 HEAT TRANSMISSION THROUGH THE TUBES OF A VACUUM EVAPORATOR

TO DETERMINE THE EFFECT OF HYDROSTATIC HEAD. WATER WAS USED IN THESE TESTS

Test Number	Duration of Test, Min.	Absolute Pressures		Temperatures, Deg. Fahr.				Height of Boiling above Lower Tube Plate, In.	Weight of Water Evaporated, Lb.	Temperature Fall, Deg. Fahr.	Water Evaporated per Sq. Ft. Heating Surface, Lb. per Hr.	Coefficient of Heat Transmission
		Vapor Space, In. Mercury	Calandria, In. Mercury	Steam Pipe	Vapor Space, Average	Calandria, Average	Juice Entering					
SERIES A—1 CALANDRIA A. NO DOWNTAKE. 56 SQ. FT. HEATING SURFACE TUBES 1⅜ IN. BY 24 IN.												
1	30	26.98	33.57	221.5	207.7	219.6	81.0	3.0	37	10.0	1.32	145
2	30	26.45	33.85	221.3	205.0	218.0	82.0	4.0	68	13.0	2.45	204
3	30	26.52	33.89	225.7	206.0	219.3	82.0	5.0	114	12.2	4.07	366
4	30	25.84	33.66	227.0	206.4	218.8	84.0	6.0	149	11.4	5.32	521
5	30	27.24	33.78	230.4	207.6	218.0	82.0	14.0	194	11.0	6.93	692
6	30	26.85	33.63	229.6	206.4	218.0	82.0	18.0	184	11.4	6.57	633
7	30	27.06	34.04	227.5	206.6	218.0	83.0	14.0	150	12.0	5.36	490
8	30	27.08	33.71	229.5	206.6	218.0	82.0	7.0	228	11.4	8.14	784
9	30	27.03	33.51	230.1	207.0	217.4	81.0	10.0	214	10.1	7.64	832
10	20	26.89	33.63	230.5	206.8	217.2	81.0	8.5	148	10.4	7.93	840
11	20	27.07	33.87	229.0	207.0	217.4	81.0	6.5	147	10.4	7.87	813
SERIES A—2 CALANDRIA C. NO DOWNTAKE. 75.4 SQ. FT. HEATING SURFACE TUBES 2 IN. BY 48 IN.												
12	20	21.80	29.93	231.4	198.6	211.0	81.5	6.0	104	12.4	4.14	365
13	20	22.04	30.01	245.5	197.2	211.65	82.0	12.0	281	14.45	11.18	844
14	20	21.93	29.81	249.4	196.8	211.7	81.0	18.0	310	14.9	12.3	908
15	20	21.96	29.93	248.7	196.8	212.15	81.0	24.0	318	15.21	12.7	912
16	30	21.76	29.90	240.0	196.4	210.8	82.5	30.0	368	14.4	9.76	741
17	30	22.21	30.04	240.8	196.6	210.9	82.8	36.0	284	14.3	7.54	576
18	30	21.74	29.90	236.1	196.2	210.4	81.5	42.0	306	14.2	8.12	605
19	30	21.80	29.88	240.8	195.6	210.85	89.0	48.0	323	15.25	8.57	609
20	30	21.91	29.92	238.6	196.1	211.15	85.0	54.0	270	15.05	7.2	517

TABLE 2—*Continued*

Test Number	Duration of Test, Min.	Absolute Pressures		Temperatures, Deg. Fahr.				Height of Boiling above Lower Tube Plate, In.	Weight of Water Evaporated, Lb.	Temperature Fall, Deg. Fahr.	Water Evaporated per Sq. Ft. Heating Surface, Lb. per Hr.	Coefficient of Heat Transmission
		Vapor Space, In. Mercury	Calandria, In. Mercury	Steam Pipe	Vapor Space, Average	Calandria, Average	Juice Entering					
Series A—3 Calandria D. 86.21 Sq. Ft. Heating Surface Tubes 2 In. by 54 In.												
21	23	20.46	29.75	241.7	194.8	219.2	70.0	6.0	300	16.8	11.8	593
22	42	22.05	30.22	228.0	198.2	212.5	70.0	12.0	940	14.15	15.23	1196
23	41	22.25	30.26	254.0	198.1	212.4	70.0	18.0	936	14.3	15.92	1257
24	27	21.45	29.80	257.2	196.1	210.2	70.0	24.0	606	14.1	15.65	1235
25	48	21.95	29.83	232.0	197.0	211.5	71.0	30.0	640	14.5	13.65	1048
26	43	21.56	30.10	255.2	195.8	211.5	71.0	36.0	894	15.7	14.49	1025
27	16	21.54	29.13	247.3	195.1	210.7	70.0	42.0	305	15.55	13.28	943
28	15	21.68	29.38	244.4	196.5	214.3	70.0	48.0	305	15.1	14.15	912
Series A—4 Calandria E. 29 Sq. Ft. Heating Surface Tubes 1¾ In. by 24 In.												
29	20	21.74	31.71	.....	196.8	216.5	80.0	2.0	193	18.0	19.98	1220
30	20	23.05	33.18	.....	199.0	219.2	78.0	3.0	226	17.9	23.45	1410
31	20	22.87	32.23	.....	198.8	220.2	78.0	5.0	228	16.8	23.55	1468
32	20	22.53	31.61	.....	198.6	219.0	78.0	6.0	214	16.0	22.14	1445
33	20	22.50	39.82	.....	198.7	229.7	79.5	12.0	312	28.0	32.25	1256
34	20	23.65	35.78	.....	199.8	225.0	79.0	15.0	226	21.2	23.4	1192
35	30	22.81	35.62	.....	199.1	225.8	79.5	18.0	337	21.85	23.22	1120
36	30	22.94	35.48	.....	200.4	223.3	78.0	21.0	270	20.1	18.63	993
37	20	22.92	32.52	.....	197.4	220.6	78.0	24.0	166	18.6	17.17	974

TABLE 3 TESTS OF HEAT TRANSMISSION THROUGH THE TUBES OF A VACUUM EVAPORATOR

TO DETERMINE THE EFFECT OF VARYING THE TEMPERATURE LEVEL. WATER WAS USED IN THESE TESTS

Test Number	Duration of Test, Min.	Absolute Pressures		Temperatures, Deg. Fahr.			Weight of Water Evaporated, Lb.	Temperature Fall, Deg. Fahr.	Water Evaporated per Sq. Ft. Heating Surface, Lb. per Hr.	Coefficient of Heat Transmission
		Vapor Space, In. Mercury	Calandria, In. Mercury	Steam Pipe	Calandria, Average	Juice Entering				
				SERIES B—1						
				CALANDRIA A. WITHOUT DOWNTAKE. 56 SQ. FT. HEATING SURFACE. HEAD 12 IN. TUBES 1⅜ IN. BY 24 IN.						
38	20	23.72	35.93	235.0	222.45	82.0	303	21.27	16.23	835
39	20	22.52	34.09	242.9	218.0	81.0	279	20.5	14.94	800
40	20	21.01	31.97	243.8	215.3	82.0	263	19.42	14.08	913
41	20	18.27	27.59	246.0	207.7	80.0	202	19.7	10.82	601
42	20	14.77	22.64	216.3	197.4	80.5	177	20.0	9.48	517
43	20	12.11	19.14	226.6	188.1	82.0	141	20.45	7.55	399
				SERIES B—2						
				CALANDRIA B. WITH DOWNTAKE. 53.26 SQ. FT. HEATING SURFACE. HEAD 16 IN. TUBES 1⅜ IN. BY 24 IN.						
44	60	29.84	35.93	241.2	222.0	83.0	494	9.2	9.28	1108
45	60	21.44	29.93	242.6	213.7	83.0	685	16.4	12.87	857
46	60	14.11	21.46	240.1	195.1	81.0	671	21.4	12.60	647
47	60	8.09	14.02	237.9	172.8	81.0	671	24.3	12.60	558
				SERIES B—3						
				CALANDRIA C. 75.4 SQ. FT. HEATING SURFACE. HEAD 24 IN. TUBES 2 IN. BY 48 IN.						
48	20	7.00	12.48	225.3	168.1	81.5	245	26.0	9.75	404
49	20	9.30	14.61	233.6	178.2	80.0	247	21.9	9.85	487
50	20	12.40	19.40	244.0	191.6	80.5	283	21.1	16.22	578
51	20	19.26	27.53	252.3	208.8	81.5	312	16.3	12.46	660
52	20	23.88	36.04	259.2	222.5	82.0	388	21.0	15.44	806

TABLE 3—*Continued*

Test No.	Duration of Test, Min.	Absolute Pressures		Temperatures, Deg. Fahr.			Weight of Water Evaporated, Lb.	Temperature Fall, Deg. Fahr.	Water Evaporated per Sq. Ft. Heating Surface, Lb. per Hr.	Coefficient of Heat Transmission
		Vapor Space, In. Mercury	Calandria, In. Mercury	Steam Pipe	Calandria, Average	Juice Entering				
				Series B—4						
				Calandria D. 86.21 Sq. Ft. Heating Surface. Head 36 In. Tubes 2 In. by 54 In.						
53	25	22.14	33.72	243.9	218.5	76.1	678	20.0	18.9	1029
54	20	17.27	26.25	245.0	205.3	76.1	428	20.45	14.85	797
55	20	12.94	19.8	229.2	192.4	76.1	395	19.8	13.74	700
56	20	9.45	14.92	230.8	179.7	76.1	225	20.1	11.65	628
57	30	6.25	10.81	239.1	165.5	73.5	551	23.3	12.84	594
58	30	5.64	9.94	236.4	161.7	73.5	545	21.25	12.65	583
59	20	5.30	8.38	229.4	154.8	76.0	273	19.4	9.5	527
				Series B—5						
				Calandria E. 29.0 Sq. Ft. Heating Surface. Head 12 In. Tubes 1¾ In. by 24 In.						
60	30	4.1	7.6	.....	150.9	79.0	224	22.44	15.45	738
61	30	4.11	7.99	.....	154.05	79.0	243	24.60	16.74	725
62	30	6.22	11.12	.....	169.15	79.0	261	23.20	18.05	830
63	25	8.28	15.05	.....	181.6	79.0	250	25.7	20.70	857
64	25	8.04	15.84	.....	189.0	77.5	353	28.75	29.27	1045
65	30	8.68	16.66	.....	189.4	77.0	390	28.60	26.80	992
66	30	9.67	18.62	.....	194.1	77.0	409	29.60	28.20	1014
67	25	10.46	20.66	.....	196.6	79.0	363	30.5	30.5	1056
68	20	13.57	27.48	.....	212.2	78.0	327	32.28	33.8	1110
69	25	19.28	32.27	.....	219.2	77.5	451	24.80	27.6	1202
70	30	23.59	34.75	.....	220.8	80.0	300	18.65	20.7	1215
71	20	23.46	35.25	.....	224.4	79.0	255	20.40	26.2	1358
72	25	20.78	35.43	.....	225.0	77.5	387	26.5	31.3	1270
73	20	22.50	39.84	.....	229.7	79.5	312	28.0	32.25	1256
74	30	22.46	40.55	.....	230.28	80.0	511	29.15	35.27	1318
75	30	23.01	43.10	.....	233.6	79.0	515	30.8	37.6	1340
76	20	23.30	43.57	.....	234.7	79.0	415	31.7	43.0	1488
77	20	22.32	47.65	.....	238.3	78.0	670	38.2	46.25	1330

TABLE 4 TESTS OF HEAT TRANSMISSION THROUGH THE TUBES OF A VACUUM EVAPORATOR

TO DETERMINE THE EFFECT OF AIR IN THE HEATING STEAM. WATER USED IN THESE TESTS

Test Number	Duration of Test, Min.	Absolute Pressure		Temperature, Deg. Fahr.					Weight of Water Evaporated, Lb.	Temperature Fall, Deg. Fahr.	Temperature above or below Saturation in Calandria	Water Evaporated per Sq. Ft., Lb. per Hr.	Coefficient of Heat Transmission	$\frac{U_n}{U_1}$
		Vapor Space, In. Mercury	Calandria, In. Mercury	Saturation Due to Pressure in Calandria	Steam Pipe	Vapor Space	Calandria, Average	Juice Entering						
						SERIES C								
				CALANDRIA A. NO DOWNTAKE. 56 SQ. FT. HEATING SURFACE. HEAD 12 IN.										
78	20	8.06	17.64	186.5	244.0	151.8	186.6	80.0	357	34.8	0.1	19.1	594	1.0
79	20	7.94	17.61	186.4	242.6	149.0	186.4	80.0	345	37.4	0.0	18.5	533	1.0
80	20	7.91	17.51	186.2	238.5	151.0	186.0	80.0	311	35.0	− 0.2	16.7	510	0.997
81	20	8.08	17.56	186.3	236.1	152.2	185.0	80.0	274	32.8	− 1.3	14.7	464	0.973
82	20	8.28	17.57	186.32	230.2	153.4	183.0	81.0	277	29.6	− 3.32	14.85	487	0.931
83	20	8.07	17.64	186.5	231.8	152.3	182.5	80.0	242	30.2	− 4.0	12.96	408	0.916
84	20	9.03	17.62	186.45	234.7	152.2	182.3	80.0	228	30.1	− 4.15	12.2	386	0.913
85	20	8.18	17.61	186.4	227.8	152.0	182.1	84.0	180	30.1	− 4.3	9.64	300	0.909
86	20	8.07	17.68	186.6	225.0	154.0	177.7	81.0	167	23.7	− 8.9	8.95	296	0.822
87	20	7.82	17.57	186.32	226.8	150.3	176.7	80.0	177	26.4	− 9.62	9.49	284	0.811
88	20	8.07	17.62	186.45	218.5	152.4	173.0	81.0	152	20.6	−13.45	8.14	258	0.742
89	20	8.19	17.62	186.45	217.8	153.0	173.1	80.0	139	20.1	−13.35	7.45	241	0.744
90	20	8.06	17.59	186.37	215.6	152.6	171.9	80.0	128	19.9	−14.47	6.86	215	0.724
91	20	7.79	17.68	186.5	211.6	152.0	171.9	81.0	144	19.9	−14.7	7.71	240	0.721
92	20	7.97	17.60	186.4	206.2	152.0	168.9	81.0	107	16.9	−17.5	5.73	179	0.676
93	20	8.04	17.57	186.32	203.7	152.0	168.3	80.0	86	15.8	−18.02	4.61	147	0.668

TABLE 5 BOILING TEMPERATURE OF SUGAR JUICE

TO DETERMINE THE EFFECT OF THE DENSITY OF THE LIQUID UPON ITS BOILING TEMPERATURE

Test Number	Absolute Pressure		Density Juice, Deg. Brix	Temperature, Deg. Fahr.			
	Vapor Space, In. Mercury	Calandria, In. Mercury		Boiling Juice	Boiling Corrected to a Pressure of 5.9 In. Mercury	Corresponding to Pressure of Boiling	Boiling above Saturation
			SERIES D—1				
			CALANDRIA E. 29 SQ. FT. HEATING SURFACE. HEAD 12 IN.				
94	22.62	39.70	18.58	199.5	143.4	198.2	3.4
95	8.74	19.71	18.58	157.9	142.9	155.9	2.9
96	9.74	20.47	21.59	163.0	143.7	160.2	3.7
97	10.02	22.25	28.61	164.6	144.1	161.6	4.1
98	14.72	27.03	29.61	180.5	144.0	178.3	4.0
99	13.9	24.89	29.62	177.8	143.7	175.7	3.7
100	7.70	18.75	32.84	154.0	144.0	150.7	4.0
101	6.62	16.49	35.86	149.2	144.8	144.7	4.8
102	28.92	40.50	42.92	214.1	146.2	210.3	6.2
103	4.70	8.80	48.80	138.0	146.8	131.5	6.8
104	4.70	9.20	50.00	139.0	148.1	131.5	8.1
105	3.73	10.60	51.00	130.9	147.7	123.0	7.7
106	29.18	42.35	51.07	215.0	147.1	210.5	7.1
107	16.71	29.21	64.0	193.0	151.6	184.0	11.6
108	4.55	13.2	70.00	143.0	153.05	130.25	13.05
109	4.25	13.13	72.00	141.5	153.85	127.75	13.85

TABLE 6 TESTS OF HEAT TRANSMISSION THROUGH THE TUBES OF A VACUUM EVAPORATOR

To determine the Effect of Varying the Density of the Boiling Liquid upon the Heat Transmission

Test Number	Duration of Test, Min.	Absolute Pressure		Temperature, Deg. Fahr.			Density of Juice, Deg. Brix	Weight of Water Evaporated	Coefficient of Heat Transmission
		Vapor Space, In. Mercury	Calandria, In. Mercury	Vapor Space	Calandria, Average	Juice Entering			
		Series D–2 Calandria E. 29 Sq. Ft. Heating Surface. Head 14 In.							
	20	5.06	8.60	134.8	156.9	81.0	0	143	758
	25	4.93	8.74	135.07	156.1	81.0	34.1	143	609
	25	4.85	7.90	135.6	154.6	81.0	45.42	115	632
	20	4.89	8.44	136.2	155.2	81.0	45.42	87	526
	20	4.77	8.27	135.2	156.6	81.0	46.16	82	493
	20	5.00	8.72	137.7	158.0	81.0	54.98	90	441
	30	4.65	9.53	135.1	160.6	81.0	61.86	87	264

FOREIGN REVIEW

CONTENTS

Absorption machine 1574
Aeroplane, French views on safety of 1563
Air, compressed, machinery, testing 1563
Ball Socket Joint 1569
Boilers, competition, between various types 1577
Boilers, gas-fired, tests of 1575
Coal dust burning in a tar-oil fired boiler furnace 1577
Diesel locomotive 1571
Dynamometer, torsion 1569
Gas, blast-furnace, as boiler fuel 1575
Gas, coke-oven, as boiler fuel 1575
Gas engine, scavenging process in 1567
Gas engines, heat consumption of 1575
Gas-fired boilers, efficiency of 1575
Hammer, compressed air, efficiency of 1566
Hammer, compressed air, test of 1563
Helicopter 1563
Joint, ball socket 1569
Life saving apparatus, testing 1567
Locomotive, Diesel engined 1571
Locomobile, superheated steam, test of 1575
Patent law, German 1579
Pressure-volume diagram, compressed air tools 1563
Pumps, canalization, valves for 1568
Refrigerating machines 1574
Respiration, artificial, apparatus for, testing 1567
Resorption machine 1574
Ropes, winding, methods of testing 1578
Scavenging process in gas engines 1567
Springs, laminated, for large annular valves 1568
Surface combustion and power production 1577

Thermal locomotive 1571
Time-path diagram, compressed air tools 1564
Torsion dynamometer 1569
Torsion strength 1578
Utilization, degree of 1566
Valves, group, in water pumps, dynamics of 1569
Valves, large annular, design and dynamics of 1568
Wheels, toothed, of different materials in mesh 1570

The Editor will be pleased to receive inquiries for further information in connection with articles reported in the Review. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

FOREIGN REVIEW

During the past month the Third International Congress of Refrigeration was held in this country. In an early issue of The Journal will be given an account of the technical sessions, with special reference to problems relating to the liquefaction of air, standardization of definitions, and the problem of heating.

THIS MONTH'S ARTICLES

In this month's issue will be found a statement of the principles on which the Altenkirch resorption refrigerating machine is based, which is of interest because it forms an essential part of a system of heating, adapted for use in large buildings, in which ice or cold is produced as a by-product, and which, as Lord Kelvin proved theoretically more than half a century ago, widens the range of temperatures utilized. A part of the potential energy of the coal is converted into heat through the efficient combination of a steam engine (with the utilization of the exhaust steam), a compressor, degaser and resorber. The system gives *an apparent* efficiency of more than 100 per cent if the number of calories and frigories obtained at the heating and cooling ends of the system be compared with the number of calories put in in the form of coal. While it does not of course contradict the law of the conservation of energy, it should receive attention both because of the increasing demand for refrigeration in the summer, and of its high thermal efficiency and economy.

The attention of designers and users of compressed air machinery is called to the article of Groedel (section Air Machinery) which gives a complete analysis of the operation of a compressed air hammer, and describes a process for testing compressed air tools by which a pressure-volume diagram may be obtained. In the same section will be found a brief article on testing apparatus for artificial respiration, and life-saving apparatus for firemen and mine-rescue stations.

The operation of water pump valves, with particular regard to

large canalization pumps is analyzed in the article of Schoene (section Mechanics) where the author shows that, if properly designed, large valves of comparatively small mass may be used quite successfully. In the same section is to be found an article on the pitch of toothed wheels of different materials. The Diesel engine locomotive, the first thermal locomotive in operation, is described and illustrated. Unfortunately no data as to its actual performance can as yet be reported.

In the section on Steam Engineering tests of the efficiency of boilers fired with blast-furnace and coke-oven gas are reported, important both on account of the scarcity of reliable data as to these kinds of steam generators and because the efficiency established is considerably higher than was often assumed. Data on the heat consumption of gas engines are also given. In the same section are reported data of tests from a publication issued by the manufacturers of a 160-h.p. superheated steam locomobile, showing an actual steam consumption of 11.4 lb. per h.p.-hr. An interesting report of the Association for the Inspection of Boilers in Dortmund, Germany, mining district, is also abstracted, and contains, among other things, data as to burning coal-dust in a tar-oil fired furnace under a fire-tube boiler. It contains also some significant references to surface combustion as likely to bring about a revolution in the entire field of power generation.

Articles on torsional strength, torsion indicators, projected new German patent law, possibilities of helicopters as a solution of the problem of safe flight, etc., are to be found in the various sections.

ABSTRACTS OF ARTICLES

Aeronautics

WALL OR BLIND ALLEY? (*Mur ou impasse?* G. Plaisant. *L'Aérophile*, vol. 21, no. 16, p. 364, August 15, 1913. 3 pp., 5 figs. *dg*). The author tries to explain the feeling of disparagement which appears to prevail in French military aviation circles, where officers are killed in practice flights without any apparent gain, and where an opinion appears to have gained ground that the present-day aeroplane has reached the highest state of perfection possible, and that unless some radical changes in construction are made, will always remain unstable and dangerous. The author maintains that such radical changes in design are possible, but that the most promising road at the present time lies in coming back to the idea of a helicopter, i. e., apparatus which possesses a lifting power independent of its speed, and points to the success, very partial of course, of the Breguet helicopter, which he describes and illustrates, as the road to follow.

Air Machinery

EXPERIMENTAL AND THEORETICAL INVESTIGATION OF COMPRESSED AIR MACHINERY (*Experimentelle und theoretische Untersuchungen an Pressluftwerkzeugen*, E. Groedel. *Werkstattstechnik*, vol. 7, no. 15, p. 462, August 1, 1913. 5 pp., 7 figs. *e4*). The first more or less complete investigation of the processes occurring in compressed air tools belongs to Harm (cp. *The Journal*, April 1913, p. 695), who was able, however, to obtain only the piston path diagram. The author describes a method worked out by him at the Technical High School at Darmstadt, Germany, permitting him to obtain the pressure-volume diagram also.

This testing installation is shown in Fig. 1A and B. The plant (Fig. A) comprises a single-cylinder multistage compressor of 120 cbm (4236 cu. ft.) of air per hour, and of a boiler plant. The compressor working at a normal pressure of 7 atmospheres effective, takes about 14 h.p. which is supplied to it by a belt drive from a 24-h.p. gas engine. The compressed air passes into a reserve tank of 3.6 cbm (127 cu. ft.) capacity, and from there through pressure reducing valves, into measuring tanks. Between the latter and the hammer hose are placed an air meter and a dashpot tank. By means of the pressure reducing valve it is possible to maintain the pressure in K_1 and K_2 constantly at 0.005 atmospheres. A mercury partial pressure manometer is conveniently used for pressure measurements.

The testing stand proper is shown mainly in B. Two iron plates P_1 and P_3 are rigidly connected with a concrete block 0.8 m (say 31.4 in.) high. By means of the bow B the hammer, lying in a boring in P_1, is drawn against P_1, and in so doing, it leans with one of its lower edges against plate P_2. P_2 is free to move, and is kept away from P_3: the snap head die D

passes through P_2; during the tests lead wedges of a special form were placed between it and P_3. For the taking of the piston path diagram, a silver thread 3 mm (0.118 in.) in diameter was appropriately fastened to the piston, led through short drilled holes of the snap head die, and provided with a steel recording apparatus of special construction. Fig. B shows a time-path diagram of a striking compressed air hammer. The wavy line to the right on the drum is a record made by a tuning fork, by means of which the peripheral velocity of the drum may be measured. The pressure was recorded by a somewhat modified Schulz optical indicator.

In Fig. C the time-path diagrams are placed below one another. As only

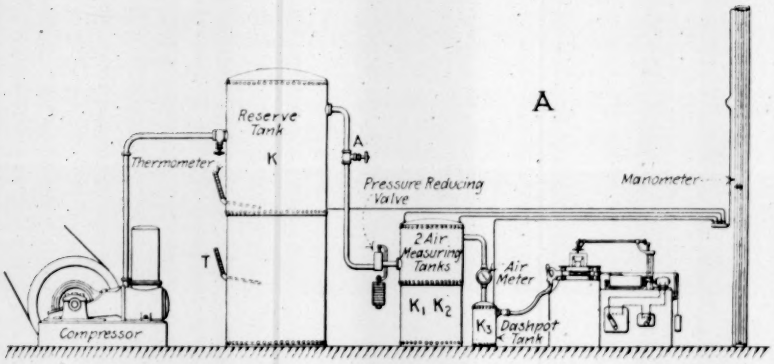


FIG. 1A PLANT FOR TESTING COMPRESSED AIR TOOLS

one indicator was available, the striking and the return blows were indicated separately, but the uniformity with which these hammers worked made the juxtaposition of such two observations appear permissible. The piston-path diagrams, as can be seen, nearly coincide. From the time diagrams the pressure-volume diagrams are derived by means of, e. g., plotting the pressure p from the time diagram for the rear side of the piston over the atmospheric line on the right side in the piston position s. The diagram obtained in this way, i. e., point after point, requires, however, some additional drawing. The spring scale of the indicator varies considerably, and from its calibration curve, corrections are obtained, lines r-r, by the use of which the final P-V diagram, drawn to a uniform scale of 1 cm.-1 atmosphere, is obtained.

For the diagram C one of the best hammers obtainable was used. From the position of the ports A, R, U and C, as indicated, it may be seen that the admission line in the diagram for the rear side of the piston is depressed, owing to throttling at entrances down to point 2, and at point 4 escape of air begins, which continues also during the return stroke of the piston, but stops when port A is closed. In the return stroke compression begins at 5, and at 6 the valve opens the admission which continues at a pressure above the working pressure until the beginning of the line of cut-off at the reversal of the stroke. During the striking blow a slight rise of pressure takes place under the piston up to 7, while the

admission lead begins. The piston is then driven back, past 8 to 9, under a fairly uniform pressure and at 9 the exhaust under the piston takes place. The author does not discuss the nature of the oscillations which appear in the diagram, but states that they are of exceptional interest for the determination of the behavior of the air and valves, and will be investigated separately.

Diagram C is used in the following way for the derivation of numerical characteristics of a compressed hammer. The velocity of the piston

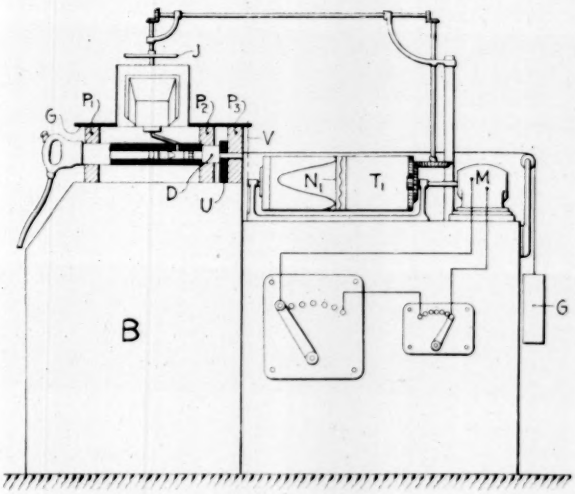


FIG. 1B TESTING STAND FOR TESTING COMPRESSED AIR TOOLS

V_e is determined by graphic differentiation by means of a tangent drawn at the end of the stroke curve and is

$$V_e = \frac{ds}{dt} = \frac{0.164}{0.01287} = 12.78 \text{ m/sec.}$$

The weight of the piston being $G = 0.612$ kg., the effective force of blow is

$$A_e = \frac{m}{2} V_e^2 = \frac{0.612}{2 \times 9.81} \times 12.78^2 = 5.1 \text{ mkg.}$$

The duration of blow t_e is determined by measurement (in this case 223 mm) on the drum circumference, or on the diagram sheet flattened out. As the tuning fork curve has the ratio 46.5 mm = 0.01287 sec.

$$t_e = \frac{0.01287 \times 223}{46.5} = 0.0617 \text{ sec.}$$

from which the number of blows per minute n_e is easily determined. Hence the hammer output is

$$L_e = \frac{A_e}{60} \times \frac{n_e}{75} \text{ 1.1 h.p.}$$

The data as to air consumption permit the derivation (it is not stated what process was used by the author) of the thermal efficiency η_t referred to adiabatic expansion of air, and the economic efficiency η_w referred to the compressor work theoretically required. In this case $\eta_t = 0.302$, and $\eta_w = 0.196$

if η_w be calculated on the basis of two-stage adiabatic compression. Practically, however, the power required at the compressor shaft is larger, and if a total efficiency of 75 per cent be assumed in the generation of compressed air, then

$$\eta = \eta_w \times 0.75 = 0.147$$

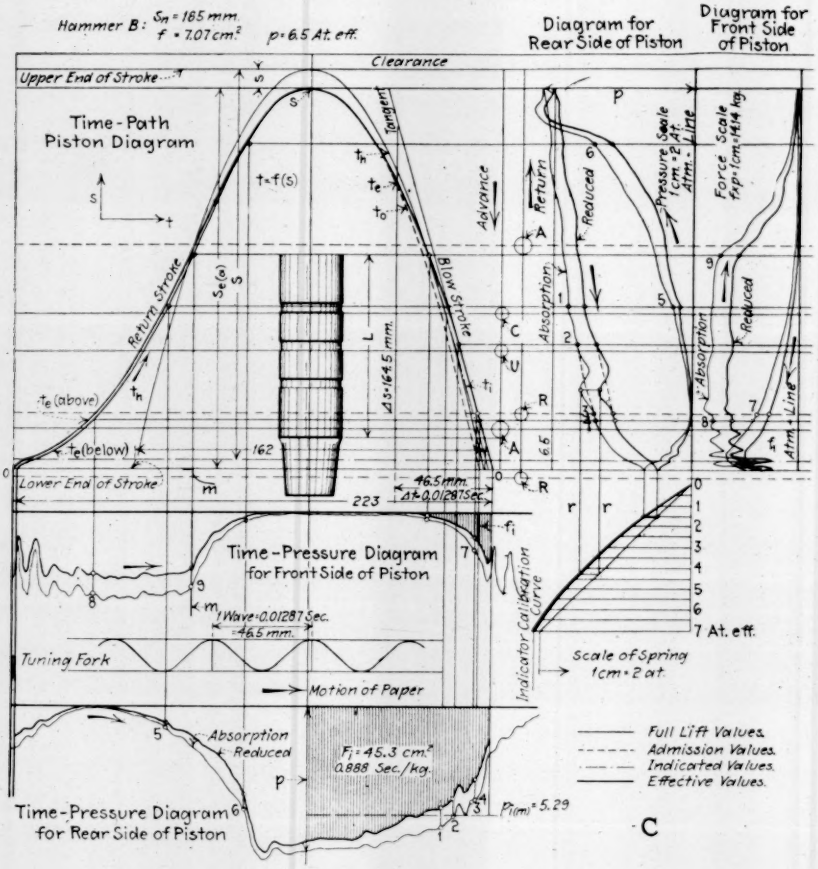


FIG. 1C DIAGRAM FOR COMPRESSED AIR HAMMER

which means that the engine driving the compressor feeding a hammer of 1.1 h.p. must deliver

$$\frac{1.1}{0.147} = 7.49 \text{ h.p.}$$

which may be increased still further through presence of untight places and consequent air leaks.

A further basis for comparing hammers of different constructions is suggested by the author in what he calls "degree of utilization" (*Ausnut-*

zungsgrad). An ideal case is assumed of a piston free from friction during the striking and return strokes with no loss of lift, driven by full and constant working pressure; for this case the force of the blow and number of blows are calculated. The striking body follows the laws of uniformly accelerated motion, and the "theoretical duration of blow" of a simple stroke is

$$t = \sqrt{\frac{2S}{b}} = \sqrt{\frac{2Sm}{f.p}}$$

where S is the "constructive" stroke, b acceleration of mass m due to a constantly acting force, while $f \times p$ means piston area times pressure. The "theoretical number of blows per minutes" is $n = \frac{60}{t}$, and the "theoretical force of blow per stroke" $A = f.p.S$. The "theoretical output" is then $L = \frac{An}{60}$ and the "degree of utilization" is the ratio

$$\eta a = \frac{L_e}{L} = \frac{1.1}{3.98} = 0.436$$

The author shows also how to determine the mechanical efficiency of the hammer, and gives tables of numerical data obtained in his tests which cannot be reported here owing to lack of space.

Devices for Testing Apparatus for Artificial Respiration (*Prüfungsvorrichtungen für Sauerstoff-Atmungsgeräte*, Forstmann. *Glückauf*, vol. 49, no. 31, p. 1216, August 2, 1913. 3 pp., 4figs. *d*). Several occasions when life-saving apparatus using artificial respiration have failed to work properly have shown the necessity of some reliable and rapid method of testing them. This applies particularly to the so-called automatic apparatus, i.e., those having an automatically working pressure-reducing valve and injector, which are liable to get out of order even with small amounts of dirt in the working parts. The author describes two special testing devices made by two concerns manufacturing the automatic life-saving apparatus. He states, however, that for rapid tests just previous to using, the special vacuum gage (*Depressionsmesser*) now in use is still the best device in existence. Tests ought to be made, however, for suction and not compression, because in this way the air-tightness of the apparatus may be also tested. Particular attention must be paid to the state which the rubber parts are in. (This applies especially to apparatus used in tropical countries where rubber articles do not last long. Editor.)

Internal Combustion Engines

Concerning the Scavenging Process of Gas Engines (*Über Ausspülverfahren bei Gasmaschinen*, A. Nolte. *Stahl und Eisen*, vol. 33, no. 32, p. 1301, August 7, 1913. 6 pp., 11 figs. *et*). Discussion of the scavenging process as a means of improving the efficiency of the large gas engine. The author describes briefly the introduction of the scavenging process, and the chief types of engines in which it is used, and cites data of some tests showing the higher efficiency of engines with scavenging as compared with those without.

Mechanics

TESTS WITH LARGE ANNULAR VALVES OPERATED BY LAMINATED SPRINGS FOR CANALIZATION PUMPS, AND CONTRIBUTION TO THE DYNAMICS OF VALVE MOTION (*Versuche mit grossen durch Blattfedern geführten Ring ventilen für Kanalisationspumpen, und Beitrage zur Dynamik der Ventil bewegung*, Kurt Schoene. *Zeits. des Vereines deutscher Ingenieure*, vol. 57, no. 32, p. 1246, August 9, 1913. (1/4). The author states that notwithstanding the large amount of experimental work already done (of which a historical sketch is given), the working processes of water pump valves, their opening and especially their closing, are as yet by no means clear. The article is not suitable for full abstracting, and therefore only the conclusions will be reported here. It is not permissible to assume, for the beginning and end of the valve motion, that the clearance velocity is constant: it actually increases, under all circumstances, from zero up to a certain value.

Valve closing, with particular reference to the influence of frictional resistance in the valve guide, and when several valves work in groups. The author states that the closing of the valve after the reversal of stroke has been hitherto treated either under the inadmissible assumption that the velocity of discharge is constant, or in a very general manner. As a matter of fact, however, what happens after the reversal of stroke is of great importance to the proper working of a pump, and the author shows that the closing of a valve (say, suction valve) occurs in the following manner: At the instant of the reversal of stroke the water column above and below the valve is at rest (aside from eddies and motions of oscillatory character), and the water which is still flowing out with considerable velocity along the circumference of the valve, fills the space above the valve which becomes free, owing to the motion of the valve. The velocity of the valve is then gradually decreased owing to the reduction of the spring action, but the partial vacuum increases, since, owing to the seats coming closer together, a greater pressure is necessary for the production of the velocity of discharge. As the displacement of the valve decreases, the influence on the closing process of the displacement of the piston increases. Because the free space above the valve is, to a large part, filled by water driven in by the piston, there is scarcely any room for water flowing over the circumference of the valve, and consequently the velocity of such flow rapidly decreases, and the kinetic energy of the column of water between the valve seats is converted into pressure acting on the valve from above, since, as a further consequence of the decrease of the velocity of discharge, the pressure under the valve decreases also, and a strong acceleration of the valve takes place, lasting down to the closing of the valve.

The above describes what happens normally when the load on the valve is large enough to communicate to it sufficient velocity and when there is no interference from special frictional or other resistances. If these conditions are not satisfied, the valve velocity rapidly decreases, and the time is reached when the piston displacement exceeds the valve displacement. At that instant the direction of clearance velocity is reversed, and water begins to flow in from outside, the pressure above the valve rises, and both the valve motion and the flow of the liquid inward are acceler-

ated. This acceleration and also the valve knock are the stronger the greater the piston velocity.

In all these processes the relations between the piston displacement and the valve displacement are of great importance. It is of advantage if the former is, up to the time of closing, always larger than the latter. The occurrence of particularly unfavorable conditions is especially apt to take place with group valves. If in these valves the guides are not free from friction, a uniform closing is not to be expected and, if out of a large number of valves, a few are kept open, the piston displacement materially exceeds the valve displacement even at high valve velocities, and the water flows violently back through the valves which are late in closing, causing strong knocks. This is the reason why group valves, notwithstanding their moderate mass, have given so little satisfaction, while large and heavy annular valves have proved both more reliable and regular

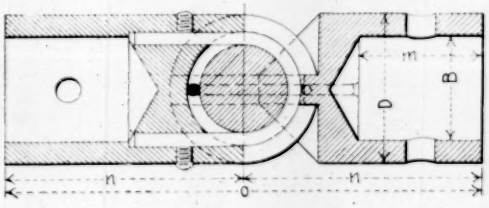


Fig. 2 Ball Socket Joint

in operation. In group valves it is important to see that the closing be as uniform and simultaneous as possible. This is helped by making the guides as frictionless as possible, and selecting large units and the lowest possible number of independent parts.

The tests (full data in the original article) show that a large valve of comparatively small mass may work quite well, if the valve discs be provided with proper laminated springs. By these means valves may be obtained of such great diameters and such long strokes that they may be used in canalization pumps instead of flap valves. Tests made in the laboratories of the Technical High School at Charlottenburg have shown the reliability of annular valves of such unusual dimensions as 300 mm (11.8 in.) average clearance diameter, and 76 mm (say 3 in.) clearance width at 60 or 70 r.p.m.

Ball Socket Joint (*Articulation à rotule*, R. d'Estrel. *Revue Industrielle*, vol. 44, no. 2008/33, p. 454, August 16, 1913. ½ p., 2 figs. *d*). Description of a ball socket joint made by the Ludwig Loewe Company of Germany. In this system (Fig. 2) the ball socket is spherical, and there is no ⊥ shaped break in the section of the ball. While joints of usual construction broke under a static load of 74 kg. (162.8 lb.), this joint resisted up to 110 kg. (232 lb.).

Torsion Dynamometer with Optical Index (*Torsions-Dynamometer mit optischer Ablesevorrichtung*, V. Vieweg. *Archiv für Elektrotechnik*, vol. 2, no. 2, p. 49, 1913. 3 pp., 2 figs. *d*). Description of a torsion dynamometer of great precision, with an indexing device constructed in accord-

ance with the Brodhun method (a rotating mirror making an angle of 45 deg. with the axis, is placed at such a distance that the virtual image formed by the nonius and scale falls into the axis of rotation). The Brodhun method has been widely applied in photometry, and with torsion dynamometers gives better results than the formerly used stroboscopic method.

PITCH OF TOOTHED WHEELS OF DIFFERENT MATERIALS IN MESH (*Teilung für zusammenarbeitende Zahnräder aus verschiedenem Material*, R. Krell. *Die Fördertechnik*, vol. 6, no. 7, p. 164, July 1913. 2 pp., 4 figs. *tA*). Some engineering handbooks give formulae for the determination of the pitch of teeth o wooden and cast-iron (mainly) wheels in mesh, but a general formula become necessary in view of the variety of materials now used, and high speeds em

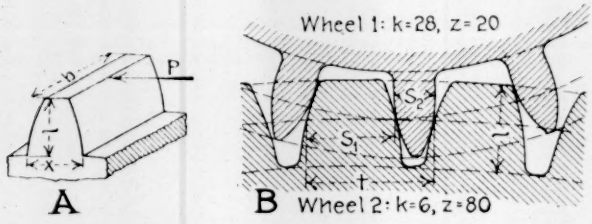


FIG. 3 WHEELS OF DIFFERENT MATERIALS IN MESH

ployed. The author gives the following solution of this problem. The general formula for the determination of pitch is (Fig. 3A):

$$P \cdot l = \frac{1}{6} b \cdot x^2 \cdot k_b$$

where k_b is the permissible bending stress on the material of the toothed wheel. If the depth of tooth l and thickness of tooth at root x be expressed in terms of pitch t, by stating that $l = \alpha t$, and $x = \beta t$, then

$$P = \frac{1}{6} \frac{\beta^2}{\alpha} k_b \cdot b \cdot t \cdot = k \cdot b \cdot t \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

The coefficient $k = \frac{1}{6} \frac{\beta^2}{\alpha} k_b$ is selected in accordance with the assumption that $1 = 0.7\ t$, and $x \cong 0.5\ t$. The required common pitch for the toothed wheels 1 and 2 (Fig. 3B) is t, the corresponding width b, depth of tooth l. The usual pitch t_1 for the wheel 1 for width b is determined from the equation $P = k_1 \cdot b \cdot t_1$, and that for wheel 2 from equation $P = k_2 \cdot b \cdot t_2$, while the common pitch is found from the equation

$$t = \frac{t_1 + t_2}{2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [2]$$

These equations, however, have not yet taken into consideration the fact that in wheel 1 the depth of tooth is somewhat greater than normal ($> 0.7\ t_1$), while in the wheel 2 it is somewhat less than normal ($< 0.7\ t_2$). This affects the coefficients k_1 and k_2 in the expressions for P. From equation [1] it appears that

$k=\frac{m}{a}$, if $m=\frac{1}{6}\beta k_b$, and the unusual depth of tooth is taken care of by multiplying the corresponding coefficient by ζ

$$'k=\frac{m}{a\,.\,\zeta}=\frac{k}{\zeta}$$

Hence $'k_1=k_1\,.\,\frac{1}{\zeta_1}$ and $'k_2=k_2\,.\,\frac{1}{\zeta_2}$

But $1=at=a\,.\,\zeta_1 t_1=a\,.\,\zeta_2 t_2$, and therefore $\zeta_1=\frac{t}{t_1}$, and $\zeta_2=\frac{t}{t_2}$

Hence

$$'k_1=k_1\frac{t}{t_1} \quad \ldots\ldots [3]$$

$$'k_2=k_2\frac{t}{t_2} \quad \ldots\ldots [4]$$

If now the two equations for P with the new coefficients be solved for t_1 and t_2 respectively, and added, with the substitution of the values of the coefficients obtained from [3] and [4], b being replaced by $b=\psi t$, equation [2] will assume the following form:

$$t=\frac{1}{2}\left(\sqrt{\frac{P}{\psi\,.\,k_1}}+\sqrt{\frac{P}{\psi\,.\,k_2}}\right) \quad \ldots\ldots [5]$$

while through further transformation the following is obtained:

$$P=4\psi t^2\frac{k_1\,.\,k_2}{k_1+k_2+2\sqrt{k_1 k_2}} \quad \ldots\ldots [6]$$

$$t=\frac{1}{2}\sqrt{\frac{P}{\psi}\cdot\frac{k_1+k_2+2\sqrt{k_1\,.\,k_2}}{k_1\,.\,k_2}} \quad \ldots\ldots [7]$$

the widths of the teeth s_1 and s_2, from equation $s=0.5\,t$, may be expressed in terms of t as follows:

$$s_1=t.\ \frac{\sqrt{k_2}}{\sqrt{k_1}+\sqrt{k_2}} \quad \ldots\ldots [8]$$

$$s_2=t\,.\,\frac{\sqrt{k_1}}{\sqrt{k_1}+\sqrt{k_2}} \quad \ldots\ldots [9]$$

Somewhat simpler equations may be deduced (the author does it) for the case when the unusual depth of tooth may be neglected, and where the bending stresses in the two materials are not very different from one another.

Railroad Engineering

THE FIRST THERMAL LOCOMOTIVE (*Die erste Thermo-Lokomotive*. F. Stornenberg. *Zeits. des Vereines deutscher Ingenieure*, vol. 57, no. 34, p. 1325, August 23, 1913. 6 pp., 20 figs. *d*). Description of the first Diesel engine locomotive. It was built for the Thermo-Locomotive Company by Sulzer Brothers in Winterthur (engines and equipment), and A. Borsig in Berlin-Tegel (frames and underframes). After preliminary tests on the line Winterthur-Romanshorn in March 1913, the locomotive was taken under its own power to Berlin, pulling on some of the sections fast freight trains, together with the steam locomotive, and developing

speeds up to 70 km/hr. (43 miles per hour), the velocity generally varying, according to schedule, from 20 to 100 km/hr. (62 miles per hour).

The power plant of the locomotive consists essentially of a driving engine direct connected with the driving axle of the locomotive and an independent auxiliary engine, having one-fifth to one-fourth the power of the driving engine, and serving for the production of compressed air, to help out the main driving engine at starting, on steep grades, etc. In addition, there are inserted between the driving and the auxiliary engines, compressed air tanks, acting as a power reserve, whether the auxiliary engine is stationary or moving. Fig. 4A schematically shows the arrangement of the thermal locomotive: *a* is the engine direct connected to the driving axle of the locomotive *b* (this engine is reversible); *c* is the auxiliary engine, serving to drive the air compressor *d* which in

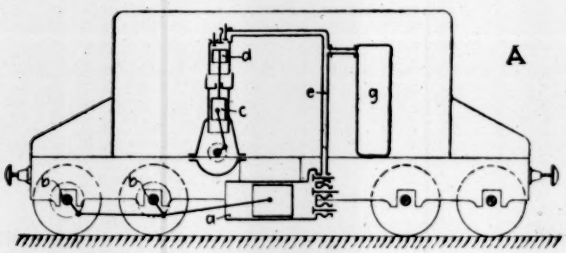


FIG. 4A SCHEMATIC ARRANGEMENT OF THERMAL LOCOMOTIVE

its turn supplies, through piping *e*, compressed air at great pressure to the driving engine *a*. At starting the auxiliary engine *c* is kept going, and supplies compressed air to start the driving engine; it can then receive comparatively large quantities of air which can be increased still further by the compressed air from the reserve tank *g*. On compressed air only, the train (?) will reach a speed of 8 to 10 miles an hour. The engine is then shifted on to fuel and goes on working with its usual regularity. Another arrangement has been patented also, in which the compressed air is supplied partly by the auxiliary and partly by the driving engine.

The general construction of the thermal locomotive is indicated in Fig. 4B. The fast train locomotive is 16.6 m (54.4 ft.) long overall, and weighs in service 95 t (104.5 short tons). The wheel base at the coupled axles is 3600 mm (141 in.), at both bogies 2200 mm (86.5 in.); the distance between bogie pins, center to center, 10,500 mm (345 ft.); the coupled wheels have tread circle diameters equal to 1750 mm (68.7 in.). The driving force is transmitted from a loose shaft to the coupled wheels by coupling rods, having their heads provided with adjustable brasses. The bogies are displaceable sidewise in order to handle freely curves up to 180 mm (590 ft.) radius. The wheels have 1000 mm (3.28 ft.) diameter. The sidewise play of the journal guides is elastically limited by powerful laminated springs acting horizontally. The coupled wheels and the laminated springs are placed under the axles. To secure the greatest possible

damping of vibrations, the suspension links for these bearing springs are themselves in the form of springs, viz., double spiral. The bogies are also on laminated springs, which apply under the axle of the bogie by means of a cradle-shaped support.

The driver's cab extends through the entire length of the locomotive. In the four corners of the carriage body are built tanks, on one end for water, and on the other for both cooling water and fuel. In the roof, over the driving engine is placed the exhaust pipe. Two pumps, driven

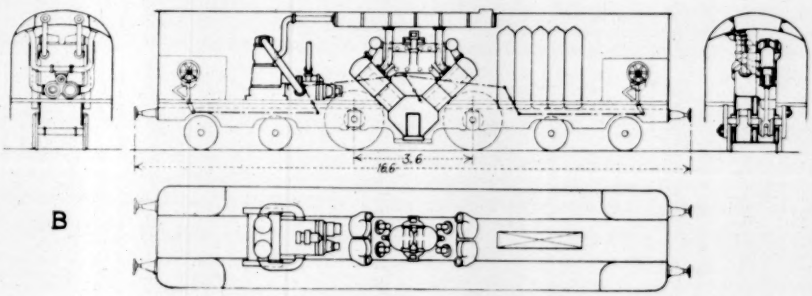


FIG. 4B GENERAL CONSTRUCTION OF THERMAL LOCOMOTIVE

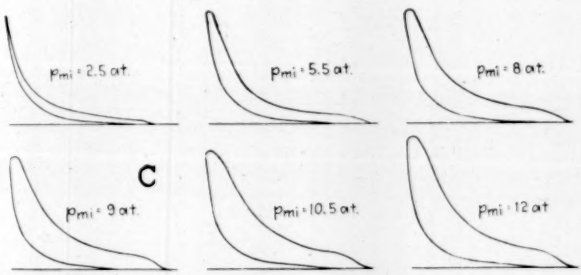


FIG. 4C INDICATOR DIAGRAMS AT VARIOUS PRESSURES

by the driving and the auxiliary engines respectively, provide for the circulation of the cooling water, suction of the water for cooling the piston, and delivery of the fuel from the fuel tank.

The driving engine is a reversible, four-cylinder, single-acting, two-stroke cycle Sulzer engine, with four cylinders arranged in pairs in V, one pair at 90 deg. to the other, and all at 45 deg. to the plane of the rails. Each pair of opposed cylinders lies in the same plane and acts on a common crank pin, the connecting rods being made forked in order to secure the most symmetrical action. The two cranks are placed at 180 deg. and make, when the locomotive runs 100 km/hr. (62 miles per hour) 304 r.p.m. The article describes in detail the very interesting arrangement for the regulation of the fuel valve and compressed air starting valve. The auxiliary engine is a 250-h.p. Diesel Sulzer, two-stroke cycle engine, having two vertical cylinders. Fig. 4C reproduces the indicator diagrams

given in the article, at various pressures. No detailed data as to tests are published as yet.

Refrigerating Machinery

REVERSIBLE ABSORPTION AND RESORPTION MACHINES (*Reversible Absorptionsmaschinen und Resorptionsmaschinen*, A. Altenkirch. *Eis-und Kälteindustrie*, vol. 6, no. 2, p. 29. August 1913. 5 pp., 5 figs. *adt*). The author discusses and compares absorption and compression refrigerating machines from the point of view of their thermal (Lorenz cycle) and economic efficiency, and indicates the conditions under which the absorption machine may have a particularly high efficiency: It must approach as far as possible the Lorenz cycle which must be closely followed, not only in the driving engine, but also in the condenser and evaporator; this can be done by effecting the liquefaction of the ammonia vapor, not by a condenser, but by absorbing it in a solution of appropriate concentration. The cold is then produced by degasification (*Entgasung*) of the highly concentrated solution. The machines are called *resorption* machines, and the author describes an older type of such a machine patented by Osenbrück. This has, however, the following imperfect details: The cold required for the precooling of the rich solution is taken from the "degaser" (as the author calls the part where degasification occurs), and therefore no improvement in the output of cold takes place through the application of the counter-current principle. Further, the liquid appears to destroy just the cold at the lowest temperature, i.e., the one most valuable thermodynamically. But if the counter-current principle be applied, then the cold in the warmer part of the absorber may be used for precoolnig the rich solution, as the cold is delivered outward at a lower temperature than when no counter-current is applied.

The same applies to the "resorber," as the absorber is called, which in these machines takes the place of the condenser. The average temperature of the resorber, as compared with that of the surrounding media, rises with the intensification of the counter-current flow of the poor solution previous to its entrance into the resorber, this being done exclusively by the counter-current flow without the expenditure of any outside work whatever. For the same *external temperature* differences as before, the application of the counter-current principle reduces the actual temperature differences between the different reservoirs, and consequently also the consumption of work. Should, however, the resorption machine be driven by an absorption machine and not by a compressor, the reduction of the actual temperature or pressure differences through the application of counter-current flow would result in a still greater overlapping of temperatures, and consequently reduction of heat consumption. This action is still greater owing to the following important fact: According to R. Plank, the output of cold of the degaser rises with the rise of temperature at which degasification occurs. As a rule this cold, obtained at a comparatively high temperature, finds no utilization, but in this case it is used for the precooling of the rich solution, which means also, in addition to increase in the absolute output of cold, an improvement in the operation of the evaporator of the ordinary absorption machine, which when pro-

vided with a condenser, may be considered as only a special case of the resorption machine. As with the absorption machine, here also, by varying the stroke volume, it is entirely possible to extend the degasification and absorption over a more or less wide reach of temperatures. If with plenty of cooling water available, it is desired to make ice, a narrow range of degasification and a wide variation of temperature are employed. Otherwise when it is desired to have the cooling water come out fairly warm, as when it is to be used as warm water, degasification is carried on largely that the poor solution may take up gas in the resorber at a comparatively high temperature.

Steam Engineering

EFFICIENCY OF STEAM BOILERS WITH BLAST-FURNACE AND COKE-OVEN GAS FIRING, AND HEAT CONSUMPTION OF GAS ENGINES (*Wirkungsgrad von Dampfkesseln mit Hochofen-und Koksofengas-Heizung und Wärmeverbrauch von Gasmaschinen*, H. Ortmann. *Stahl und Eisen*, vol. 33, no. 34, p. 1397, August 27, 1913. 3 pp. c.t.). Data of tests of gas-fired boilers.

TABLE 1A TESTS OF A COKE-OVEN GAS-FIRED BOILER (HEATING SURFACE 86QM =925 SQ. FT.)

Date of Test	Duration		Water Consumption		Average Steam Pressure		Average Gas Pressure		Gas Consumption		Efficiency Per Cent
	Hr.	Min.	Kg.	Lb.	Kg/Cm	Lb./Sq. In.	Mm/WS	In./Water	Cbm	Cu. Ft.	
Aug. 12, 1912	2	40	2238	49236	5.1	72.5	5.2	0.20	652.48	23033.	74.9
May 2, 1913	2	35	2424.5	53339	4.85	68.2	5.2	0.20	654.08	23086.	80.2

The author states that the efficiency of such boilers has been hitherto only assumed, the difficulty of measuring the amount of gas consumed being such as to make the results of tests generally unreliable. He had at his disposal a gasometer of known volume, and could therefore measure the gas consumed with the desirable degree of precision. The tests were made with a double-flue boiler, of about 90 qm (968 sq. ft.) heating surface fired by a blast-furnace gas and a similar boiler, 86 qm (925 sq. ft.) heating surface fired by coke-oven gas, both with economizers, superheaters, and Terbeck burners built into the flues.

The efficiencies obtained (Table 1A) for the coke-oven gas-fired boiler are 74.9 and once as high as 80.2 per cent. In both tests the quantity of gas was somewhat too low, so that the gas pressure was only 5.2 mm (0.20 in.); the water evaporated per hour per qm was therefore only 11.8 and 12.8 kg. (2.4 and 2.6 lb. per sq. ft. per hour). With the increase in the gas pressure, the boiler output and efficiency are expected to increase materially. Table 1B contains data of tests of a double-flue boiler with corrugated flue tubes 90 qm (968 sq. ft.) heating surface and 56 qm

TABLE 1B EVAPORATION TESTS, BLAST-FURNACE GAS-FIRED BOILER

	Test No. 1	Test No. 2	Test No. 3
Duration of test, hours	1	1	1
Gas, per cent			
Dust content	0.038 cbm = 0.4 grains per cu. ft.		
CO_2	12.4	11.4	11.4
O	....	....	....
CO	26.6	27.4	27.4
H	4.8	4.2	4.0
N	56.3	57.0	57.2
Heating value, cal	946	897	895
Temperature, deg. cent./fahr.	15/56	19/66.2	19/66.2
Air temperature, deg. cent./fahr.	18/64.4	23/73.4	23.5/74.3
Barometer, mm/ in. mercury	738/29.04	746/29.35	746/29.35
Gas pressure in piping in front of furnace, mm/ in. water	19/0.74	25/1	25/1
Gas consumption, cbm/cu. ft. per hr.	1425/50302	1515/53479	1353/47761
Feedwater			
Consumption, kg/lb	1519/3342	1280/2816	1380/3036
Temperature in front of economizer, deg. cent./fahr.	18/64.4	21/69.8	21/69.8
Same behind economizer, deg. cent./fahr	66/150.8	105/221	104/219.2
Evaporation per hour per qm/sq. ft. heating surface	16.8/180	14.2/152	15.4/165
Evaporation per qm/sq. ft. per hour of water at 0 deg. cent. (32 deg. fahr.) and steam at 100 deg. cent. (212 deg. fahr.), kg./lb.	18/39.6	14.2/31.2	15.8/34.7
	Test No. 1	Test No. 2	Test No. 3
Steam			
Average gas pressure, atmospheres	5.3	5.6	5.6
Corresponding temperature, deg. cent./fahr.	160/320	162/323.6	162/323.6
Heat of generation of saturated steam, WE per kg/B.t.u. per lb.	661/1190	662/1191.6	662/1191.6
Superheat, deg. cent./fahr.	335/635	321/609.8	343/649.4
Heat of generation of superheated steam according to Mollier J–P diagram, WE per kg/B.t.u. per lb.	752/1354	744/1339	756/1361
Evaporation			
Water evaporated, kg/lb. per kg/lb. of gas	1.02/1.02	0.85/0.85	1.01/1.01
Exhaust Gases			
Temperature, deg. cent./fahr.	285/545	290/554	295/563
Analysis			
CO_2	23.4	24.1	23.3
O	....	0.8	1.0
CO	3.4	....	0.2
Draft, mm/in. water			
In furnace	12/0.47	10/0.4	10/0.4
In chimney	20/0.8	15/0.6	13/0.507
Heat Balance, per cent			
a Steam generation	64.5	61	63
b Superheating	9.7	7.8	10.3
c Heating water in economizer	5.2	8.0	9.4
Efficiency			
Including the economizer	79.4	76.8	82.7
Excluding the economizer	74.2	68.8	73.3

TABLE 1C HEAT CONSUMPTION OF A GAS ENGINE AT VARIOUS LOADS

	Test No. 1	Test No. 2	Test No. 3
Load, kw	1920	1600	412
Load, per cent	87	73	18.8
Heating value of gas, WE/B.t.u.	976/1757	934/1681	884.6/1592.3
Gas consumption per kw/hour, cbm/cu. ft.	3.8/128.2	4.5/158.8	10.2/360.0
Heat consumption per kw/hour, WE/B.t.u.	3709/14687	4203/15644	9003/35652

(602 sq. ft.) superheater surface. The tests which are stated to have been conducted in a very exact and scientifically correct manner have shown efficiencies of 76.8, 78.4 and 82.7 per cent. The variation in efficiency was due to varying the admission of gas and air, so that, e.g., in the first test with CO content in the exhaust gases of 3.4 per cent, the efficiency was 79.4 per cent, while in the third test with only 0.2 per cent CO it rose to 82.7 per cent. On the average, however, with proper attendance, the efficiency of a gas-fired boiler may be taken to be between 78 and 80 per cent. Some sort of pressure-regulating device ought to be installed in the piping, to keep the gas pressure uniform.

The ability to measure correctly the amount of gas consumed was further used to make heat consumption tests on gas engines. Table 1C gives the data of a test on an engine driving a dynamo, the engine having 1300 mm (51.2 in.) bore, 1400 mm (55.1 in.) stroke at 94 r.p.m. rated at 3000 to 3100 effective horsepowers. Tests were made with a load on the engine of 75 to 80 per cent of full load, which is somewhat more than the usual 73 to 78 per cent.

Data of Tests of a 160-H.P. Superheated Steam Locomobile (*Versuchsergebnisse einer 160-PS-Heissdampflokomobile. Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 36, no. 35, p. 431, August 29, 1913. Taken from *Lanz's Mitteilungen über Lokomobilen*, 1913, no. 12. *e*). Data of tests made at a German porcelain button factory of a Lanz superheated steam compound locomotive used as a driving engine at the factory. It has condensation by injection, is rated at 160 h.p., but may be loaded up to 200 h.p. for a long time, and up to 225 h.p. for a brief time. Normal speed is 190 r.p.m. The data of the tests (average values) are given in Table 2.

TABLE 2 DATA OF TESTS OF A LANZ SUPERHEATED STEAM LOCOMOBILE

Duration of test, min.	377
Output on brake during test, h.p.	160.7
Admission, per cent	25.6
Mechanical efficiency, per cent	91.6
Average speed, r.p.m.	191.4
Boiler pressure, atmospheres	12
Steam temperature at entrance to high pressure cylinder,	
deg. cent.	292.4
deg. fahr.	558.3
Steam consumption per effective horsepower per hour,	
guaranteed, kg/lb.	5.3/11.66
actual, kg/lb.	5.18/11.4
Coal consumption per effective horsepower per hour at 7500 WE (B.t.u. per lb.)	
guaranteed, kg/lb.	0.58–0.61/1.27–1.34
actual, kg/lb	0.571/1.256

The guarantee applied only to coal containing not more than 5 per cent ash and slag residue.

Report of the Association for the Inspection of Boilers in the Mines of the Dortmund Mining District for the Year 1912-1913 (*Bericht des Dampfkessel-Überwachungs-Vereines der Zechen im Oberbergamtsbezirk Dortmund über das Geschäftsjahr* 1912-1913. *Glückauf*, vol. 49, no. 31, August 2, 1913. 3 pp. *g*). Abstract of the report. In the period covered

by the report the Stirling type of boiler made its appearance in the district, and the reporters believe that, while the fire-flue boiler will for some time to come maintain its leading position for certain classes of work, and particularly for plants with operation of mixed nature, the Stirling boiler will come into strong competition with the water-tube boiler. Tests to establish definitely the value of this type of boiler are now under way.

The report contains also data of tests of burning coal dust in a tar-oil fired furnace under a fire-tube boiler. The coal dust was falling into the oil flame from above. The small particles burned at once, while the larger ones fell on a special grate, and burned more slowly, with a long clear flame. Coal dust consumed per hour was 18 kg (39.6 lb.) of tar-oil and 190 kg (41.8 lb), with a steam generation of 16 kg per 1 qm (say 3.3

TABLE 3 BENDING AND TORSION STRENGTH OF WINDING ROPES

Kind of Rope	Diameter, Mm.	Bendings, Average	Minimum	Torsions	
				Average	Minimum
Good	5	21.4	18.5	46.4	39.9
Poor	...	18.3	13.7	24.2	4.3
Standard	...	....	11	....	23
	1.8	....	....	....	....
Good	...	16.5	14.8	43.6	32.3
Poor	...	15.3	14.0	25.2	11.5
Standard	...	....	9	....	21
	2.0	....	....	....	....
Good	...	13.6	12.4	34.5	26.8
Poor	...	12.9	12.0	27.5	19.5
Standard	...	....	7	....	19
	2.8	....	....	....	....
Good	...	8.0	7.1	28.3	17.4
Poor	...	6.2	5.6	12.9	1.4
Standard	...	....	4	....	11

lb. per sq. ft.) heating surface. Extensive tests are to be made also of the surface combustion, and the report points out that if the promises of 95 per cent efficiency prove true, a revolution in the power production in favor of steam may be expected in the district.

Strength of Materials

TORSION STRENGTH (*Torsionsfestigkeit*. O. Speer. *Der Bergbau*, Vol. 26, no. 33, p. 531, August 14, 1913. 1½ pp. *cc*). The author insists on the insufficiency of bending tests, and superiority of torsion tests in which the torsion is measured by the angle of twist, for determining the strength of winding ropes. A comparatively weak rope may, in bending tests, give fairly good results, but its weakness is instantly revealed by torsion tests. Table 3 gives average results obtained by the author in his tests (made on behalf of the German Winding Rope Commission), in which every figure stands for an average of ten tests.

The author states further that in his long practice of testing winding

ropes he never ran across a case where a new rope had to be rejected on account of damage which it had undergone in packing or transportation.

Miscellanea

THE PROJECT OF THE NEW PATENT AND TRADE MARK LAW (*Zu dem Entwurf für das neue Patent-, Gebrauchsmuster- und Warenzeichengesetz*, E. Bierreth. *Dinglers polytechnisches Journal*, vol. 328, no. 33, p. 513, and no. 34, p. 537. 5 pp. *d*). Exposé and criticism of the new German patent law, covering patents, limited patents (*Gebrauchsmuster*) and trade marks. The more interesting provisions of the project are as follows: The yearly payments are considerably reduced (50 marks for each of the first five years, and a progressive addition of 50 marks each following year up to the end of the life of the patent, making a total of 3500 marks, instead of 5280 marks as now). On the other hand, all the other fees are materially increased: application fee from 20 to 50 marks, appeal fee in the same proportion; anybody objecting to the grant of a patent must pay a fee of 20 marks, and is liable to pay costs if his objection is not sustained. An objection alleging lack of patentability involves a fee of 100 marks, and an appeal to the Imperial Court on that ground, a fee of 300 marks. The invention of an employee belongs to the employer, in the absence of special agreements covering this case, if the invention is in a field within the regular work of the employee; but the employee has a right to a compensation for his invention, and may insist on being named as the inventor. Patents are to be granted to the real inventor, and not the first applicant, as heretofore. The procedure in the patent office is somewhat simplified, and the period of the life of the patent is made to start from the date of the publication of the application, and not from that of the filing of the application as now. "Restoration into the original state," i.e., action after statutory date, is permitted in a number of cases, e.g., appeals, etc., but not in the case of failure to pay on time the yearly fees, in which case the lapse appears to be irreparable.

NECROLOGY

EDWARD MINER ADAMS

Edward Miner Adams was born in Pittsburgh, Pa., July 19, 1868, and went to Crestline, Ohio, when two years of age. He had only a common school education, being obliged to work in the Pennsylvania Railroad shops when thirteen years old. Here he learned boiler making and the coppersmith's trade. Later he went to Bucyrus, Ohio, where he was employed by the Thompson Steam Shovel Company as machinist. In 1892 he moved to Akron, Ohio, and worked as machinist with the Akron Machine Company. After three years he accepted an offer from the American Cereal Company, later the Quaker Oats Company, and in 1898 attained the position of machine shop foreman. In the same year he was made chief engineer in addition to his other duties, a position he retained until the time of his death on April 17, 1913.

Mr. Miner was president of the Ohio Society of Mechanical, Electrical and Steam Engineers, having been chosen to serve a second term; also a member of the Akron Chamber of Commerce.

FRED H. DANIELS

Fred H. Daniels was born at Hanover, N. H., June 16, 1853. Immediately after graduation from Worcester Polytechnic Institute he entered the employ of Washburn & Moen Manufacturing Company as draftsman and came under the direct supervision of Charles H. Morgan, general superintendent of the company. Soon afterwards he was transferred to the machine shop and in this department of the works obtained an experience which proved of infinite value in his later career as an engineer and manufacturer of wire rods and wire.

About the year 1874 his company thought it imperative that a laboratory in charge of a competent chemist be added to their works, and decided to send Mr. Daniels to study under Thomas M. Drown of Lafayette College, Easton, Pa. Here he remained 18 months, and then was sent, in company with Mr. Morgan, to study the state of the wire industry in England, France, Germany, Norway, Sweden

and Belgium. He returned to Worcester in April 1876 and again took charge of the drafting room and laboratories, where he inaugurated new and advanced practice based upon his investigations. In 1878 he accompanied Mr. Morgan abroad a second time to inspect a continuous mill with horizontal rolls in Germany. Practically all continuous mills heretofore had alternating horizontal and vertical rolls, a combination possessing great disadvantages. Upon their return to Worcester they were granted patents on the horizontal rod mill as well as on the reels, and a mill was built which proved to be very successful. The tonnage was doubled and rods two sizes smaller than had ever been rolled were produced. The invention revolutionized the art of rod rolling throughout the world. Both the reels and the mills are now almost in universal use, reducing the cost of rolling $2.50 per ton.

Shortly afterwards Mr. Daniels was promoted to the position of superintendent of buildings, retaining charge of the drafting room and laboratories. In 1887 upon the resignation of Mr. Morgan from the company, P. W. Moen was made general superintendent and Mr. Daniels assistant general superintendent of all the Washburn & Moen plants. About this time the business had grown to such proportions that it was decided to open a second plant at Waukegan, Ill. Mr. Daniels designed and superintended the construction of the buildings having an output of 800 tons per day of all kinds of wire rods and wire. After acting as assistant general superintendent for a year, he was promoted to general superintendent, and in 1895 was sent to San Francisco, where in conjunction with Frank L. Brown, Pacific coast sales agent, he established the Hallide works, known as the California Wire Works. In 1898 the Washburn & Moen Manufacturing Company was taken over by the American Steel & Wire Company and Mr. Daniels became engineer of their 30 plants. He was given the responsibility of putting a number of rundown properties into first class condition, and the business was then sold to the United States Steel Corporation. Mr. Daniels was appointed chairman of the board of engineers, in charge of 143 plants.

During 1902 and 1903 Mr. Daniels designed and constructed at San Francisco the Pacific coast works of the American Steel & Wire Company. In 1907 he was delegated to go abroad by his company to inform himself fully regarding rolling mills for producing hot rolled flats. Simultaneously the United States Steel Corporation authorized him to negotiate for the purchase of a shop right for its plants in the patented device of O. E. Theisen of Munich for purifying blast-furnace gases.

In 1907 and 1908 he designed and constructed for his company the Cuyahoga works at Cleveland, and in 1910 and 1911 their Birmingham works.

Mr. Daniels was a member of the American Institute of Mining Engineers, the American Society for Testing Materials, the Iron and Steel Institute of Great Britain, the American Iron and Steel Institute, and was a Life Member and Past Vice-President of the Society. He died August 31, 1913.

CHARLES SIMEON DENISON

Charles Simeon Denison was born at Gambier, Ohio, July 12, 1849, and was educated at the high school at Lockport, N. Y. He spent one year at Norwich University, Northfield, Vt., where he acquired a military training. In 1871 he was graduated from the University of Vermont with the degree of C.E.; in 1874 he received the degree of M.S.; and in 1907 the honorary degree of Sc.D. was conferred upon him by his alma mater.

He was assistant engineer in construction of the Milwaukee and Northern Railroad from 1871 to 1872, and since then had been instructor at the University of Michigan, rising through various positions to head of the mechanical drawing department, which he held at the time of his death on July 30, 1913.

Mr. Denison was United States astronomer and surveyor in locating the boundary line between Washington and Idaho territories in 1873 and 1874. The accuracy of his work upon this survey, accomplished largely in the winter under severe conditions and great hardships, was fully established by a recent expedition sent out by the United States Government for placing permanent markers on the boundary. He was a member of the Society for the Promotion of Engineering Education, the Michigan Engineering Society, Detroit Engineering Society, and an honorary member of Tau Beta Pi, and was the author of numerous published essays and lectures on scientific subjects. He had been an extensive traveler in America and Europe, largely in the interest of his work at the University of Michigan.

EDWARD J. MURPHY

Edward J. Murphy was born in the province of Ulster, Ireland, February 5, 1829. He was educated in the private schools of Dublin, his training being that of civil engineer. In 1849 he came to America, and the following year made surveys in Ohio for a Phila-

delphia map publishing company, and later did the same kind of work in the central part of New York. In 1853 he became first assistant to the city surveyor of New York, helping to lay out many of the street car lines of the city. Two years later he went to Hartford as chief draftsman for the Woodruff & Beach Iron Works, and although the work was somewhat different from that of a civil engineer, he gave complete satisfaction from the first, and continued with the firm until its dissolution. In this capacity he was connected with some of the most important work for the government during the Civil War, being identified in the designing and constructing of the machinery for the United States sloops of war Mohican and Kearsarge and also with the gunboats Pequot and Nipsic, and later with the three large sloops of war Piscataqua, Minnetonka and Manitou.

After the war the Nelson Mining Company was organized in Hartford and Mr. Murphy went to Montana to look after the interests of Woodruff & Beach, in the hope that the change would improve his health which had become impaired. Soon after his return the Hartford Foundry & Machinery Company was formed to succeed the old firm of Woodruff & Beach, and Mr. Murphy was elected secretary and treasurer of the company at its organization in 1872. In 1878 he was elected president of the Board of Water Commissioners, from which he resigned two years later to accept an appointment of superintending engineer at the factory of the Colt Patent Fire Arms Manufacturing Company. He remained with this firm until 1889 when he resigned to become the consulting engineer of the Hartford Steam Boiler Inspection & Insurance Company. He died September 2, 1913.

Mr. Murphy was honored with many places of public and private trust during his long residence in Hartford. He was one of the charter members of the Society and an associate member of the United States Naval Engineers' Society.

ELMER A. SAMMONS

Elmer A. Sammons was born in Cheboygan, Mich., March 31, 1860, and at the early age of 15 was made captain of the steamer Minnie Sutton, on Lake Superior. In 1880 he moved to Cincinnati, where he carried on a general consulting business for the following firms: George Enger Carriage Company, Miller Plating Works, Union Laundry Company, T. A. Snyder Preserve Works, Henry Geiershoffer Clothing Company, and W. H. Meredith and Company,

general machinists. He conducted a night school from 1885 to 1888 in connection with the Marine Engineers Beneficial Association, of which organization he was president and vice-president. In 1888 Mr. Sammons went to Louisiana where he took charge of the sugar refinery of James H. Laws & Company at Cinclare, and three years later he remodeled the entire plant, installing the revolving grate bar, one of his own patents, which proved to be so successful that it was put into practically every large factory throughout Louisiana. During this time he also superintended the erection of the A. Wilbert Refinery at Plaquemine, La. Mr. Sammons moved to New Orleans in 1903 and engaged in the general machinery business, one of his greatest undertakings being in connection with the sugar refinery at Thibodaux. He died at New Orleans, July 4, 1913.

ACCESSIONS TO THE LIBRARY

WITH COMMENTS BY THE LIBRARIAN

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A.I.E.E. and A.I.M.E. can be secured on request from Calvin W. Rice, Secretary, Am. Soc. M. E.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Trans., vol. 34, 1912. *New York, 1913.*

BRIDGES OF CLEVELAND, H. G. Tyrrell. (Reprint from The Ohio Architect, Engineer and Builder. Gift of author.

CONNECTICUT SOCIETY OF CIVIL ENGINEERS. Papers and Trans. for 1912. *New Haven, 1913.* Gift of society.

COURS D'AÉRONAUTIQUE. L. Marchis. 3 vols. *Paris, Dunod et Pinat, 1910-1912.*

These three volumes which are lithographed, as is quite usual with professorial lectures in France, are certain proof of the rapid advance in the science of aviation: ten years ago a work of 1000 pages on aviation would be impossible. The author is professor of aviation in the University of Paris.

DEUTSCHER BETON VEREIN. nos. 1-3. *Berlin.*

ELECTRIC POWER FROM THE MISSISSIPPI RIVER. (Mississippi River Power Co. Bulletin no. 10.) *Keokuk, 1913.* Gift of Mississippi River Power Co.

ELEKTRISCHE KRANAUSRÜSTUNGEN DER SIEMENS-SCHUCKERTWERKE NACH 25 JAHRIGER ENTWICKELUNG. pts. 1, 2. *Berlin, 1913.*

JOHN FITCH. The first in the world's history to invent and apply steam propulsion of vessels through water. *1912.* Gift of Francis B. Allen.

DIE FÖRDERUNG VON MASSENGÜTERN, Georg von Hauffstengel. ed. 2, vol. 1. *Berlin, 1913.*

LE FROID INDUSTRIEL, L. Marchis. *Paris, Felix Alcan, 1913.*

The author is professor in the University of Paris. The volume, although small and inexpensive, summarizes in a remarkable way the art as it exists to-day. This is the kind of book that the United States Senate contemplated taking from the free list.

GAS ENGINE HANDBOOK, E. W. Roberts. *Cincinnati, Gas Engine Publishing Co., 1913.*

This edition of a well-known handbook has been rewritten, new matter being added, specially in the discussion of frame design and in the directions for tests.

HANDBUCH FÜR EISENBETONBAU, F. von Emperger. ed. 2, vol. 9. *Berlin, 1913.*

DIE HEBEMASCHINEN, August Pohlhausen. *Leipzig, 1898.*

ILLUSTRIRTE ZEITUNG. Gas-Jahrhundert-Nummer. Entstehung des Kohlengases. Gift of Paul Gerhardt.

IMPERIAL COLLEGE OF SCIENCE AND TECHNOLOGY AND CITY AND GUILDS OF LONDON INSTITUTE. Prospectus of the City and Guilds (Engineering) College, August, 1913. *London, 1913.* Gift of Imperial College of Science and Technology.

JAHRBUCH DER WISSENSCHAFTLICHEN GESELLSCHAFT FÜR FLUGTECHNIK. vol. 1, 1912-13. *Berlin, 1913.*

LEEDS COMMERCIAL YEAR BOOK. (English, French and Spanish.) 1913. *Leeds, 1913.* Gift of Chamber of Commerce of Leeds.

MECHANICAL AND REFRIGERATING ENGINEERS' HANDY BOOK, Otto Luhr, with the collaboration of E. T. Henius. *Chicago, Wahl-Henius Institute, 1913.*

This work of about 900 pages is designed for the use of refrigerating and steam engineers. After chapters devoted to elementary physics, electricity and chemistry, it treats of furnaces, boilers, engines, power transmission, pumps, compressors, and finally, of refrigeration and its applications. Numerous useful tables and a most comprehensive index are provided.

OSTERREICHISCHER INGENIEUR UND ARCHITEKTENVEREIN. Jahrbuch 1913. *Wien, 1913.* Gift of society.

———Reform des technischen Hochschulwesens in Osterreich. April 26, 1913. *Wein, 1913.* Gift of society.

PERMANENT INTERNATIONAL ASSOCIATION OF NAVIGATION CONGRESSES. Proc. 12th congress, Philadelphia, 1912. *Brussels, 1912.*

PRACTICAL METALLOGRAPHY OF IRON AND STEEL, J. S. G. Primrose. *Manchester, England, Scientific Publishing Co.*

While primarily designed as a textbook of moderate price, this volume summarizes the processes and results of metallographic investigation in such a way as to make it a valuable addition to the library of any engineer.

PRODUCER GAS, J. E. Dowson and A. T. Larter. ed. 3. *London-New York, 1912.*

RESISTANCE OF THE AIR AND AVIATION, G. Eiffel, translated by H. C. Hunsaker. ed. 2. *London, 1913.* Gift of G. Eiffel.

This work of a distinguished honorary member of the Society makes available in English an authoritative account of the extremely valuable and interesting laboratory experiments made by him in Paris. The original French edition has been supplemented by an account of the later experiments at Auteuil, which were originally published in the Mémoires of the Société des Ingenieurs Civils.

RULES OF MANAGEMENT, WITH PRACTICAL INSTRUCTIONS ON MACHINE BUILDING, William Lodge. *New York, McGraw-Hill Book Co., 1913.*

The author is president of a large machine tool company; this book seems to have been written in chapters, each giving a general summary of the duties of some particular official or employee in such a company. The style is direct and clear cut, and the chapters short. It should be extremely valuable to every machine shop employee. Suggestions are given for the establishment of a pension system.

SAYINGS AND WRITINGS ABOUT THE RAILWAYS, by those who have managed them and those who have studied their problems. *New York, 1913.* Gift of Bureau of Railway Economics.

SCIENCE OF BURNING LIQUID FUEL, A PRACTICAL BOOK FOR PRACTICAL MEN, W. N. Best. *1913.* Gift of author.

The author has had years of practical experience in the use of fuel oil and has designed and patented numerous appliances for its economical use. The volume is largely devoted to the description of this apparatus.

THE TECHNICAL PAPERS OF ARIYA INOKUTY. *Tokyo, 1913.* Gift of the Celebration Committee.

Professor Inokuty has occupied the chair of engineering in the Tokyo University for 25 years, and the present volume is a beautiful tribute from his former pupils and from friends. It contains a reprint of his technical papers, 43 in number, of which 11 are in the Japanese language, the remainder in English. The volume is beautifully printed.

THAYER SCHOOL OF CIVIL ENGINEERING. Annual for 1909-1912. *Hanover, 1909-1912.* Gift of Dartmouth College.

THEORY AND DESIGN OF STRUCTURES, E. S. Andrews. ed. 3. *London, Chapman & Hall, 1913.*

———Further Problems in the Theory and Design of Structures, E. S. Andrews. *London, Chapman & Hall, 1913.*

These contain advanced problems for the use of students and draftsmen.

THEORY OF HEAT, Thomas Preston, revised by J. R. Cotter. ed. 2. *London, 1904.*

GIFT OF M. K. DE MEYENBURG

La motoculture par tracteurs ou machines rotatives.

Motorkultur-Maschine (system v. Meyenburg) für hochwertige, neuzeitliche Feldbestellung.

Motorlaftwagen und Motorpflüge. (Sonderabdruck "Mitteilungen der Deutschen Landwirtschafts-Gesellschaft, 1910.")

Motorpflüge und Bodenfrasmaschinen. (Sonderabdruck "Jahrbuch der Deutschen Landwirtschafts-Gesellschaft, 1912.")

Le rendement du tracteur agricole.

EXCHANGES

ENGINEERS' CLUB OF SAINT LOUIS. 18th Annual Bulletin. 1913.

INSTITUTION OF CIVIL ENGINEERS. Minutes of Proceedings. vol. 192. *London, 1913.*

WISCONSIN GAS ASSOCIATION. Proc. 12th annual convention. *Milwaukee, 1913.*

EXCHANGE ACCOUNT—UNIVERSITY OF MICHIGAN

AERONAUTICS. vol. 1, nos. 12, 13. *London, 1908.*

GRANITE CUTTERS' JOURNAL. vol. 32, nos. 10-12, 1909; vol. 35, nos. 1-12. *Quincy, Mass. 1909, 1911.*

ILLINOIS SOCIETY OF ENGINEERS AND SURVEYORS. Proc., 1899, 1903.

IOWA CIVIL ENGINEERS AND SURVEYORS SOCIETY. Proc., 1895, 1896.

OHIO SOCIETY OF SURVEYORS AND CIVIL ENGINEERS. 16th, 17th annual report, 1895-1896.

THE TECHNOGRAPH. vols. 11, 15-19. *Urbana, 1897, 1901-1905.*

THE TRANSIT. vols. 5, 7. *Iowa City, 1897, 1899.*

VALVE WORLD. vol. 1, no. 5; vol. 3, no. 11. 1905, 1907.

ZEITSCHRIFT FÜR DAS GESAMTE TURBINENWESEN. Yr. 3, nos. 1-2, 4-6, 8-11, 15-36. *Berlin, 1906.*

UNITED ENGINEERING SOCIETY

NEW YORK TIMES INDEX. January-June (2 vols.). *New York, 1913.*

LES NOUVEAUX LIVRES SCIENTIFIQUES ET INDUSTRIELS. vols. 1-2, 1902-1912. *Paris, 1908, 1913.*

PROGRESS OF GERMAN SHIPBUILDING, WITH SPECIAL REFERENCE TO THE EVOLUTION OF THE FLEET OF THE NORDDEUTSCHER LLOYD. (English-German.) *Berlin, 1909.* Gift of North German Lloyd Steamship Co.

TROW'S GENERAL DIRECTORY OF THE BOROUGHS OF MANHATTAN AND BRONX, CITY OF NEW YORK. vol. 127, 1913. *New York, 1913.*

TRADE CATALOGUES

EDGE MOOR IRON CO., *Edge Moor, Del.* Bull. 51, tests of a steam boiler at high rates of evaporation made at the Westport Generating Station Consolidated Gas Electric Light and Power Co. of Baltimore, 1913.

ASBESTOS PROTECTED METAL CO., *Beaver Falls, Pa.* Bull. 53, asbestos steel for roofs and walls, 1913.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing positions for its members. The Secretary gives this his personal attention and is pleased to receive requests both for positions and for men. Notices are not repeated except upon special request. Names and records, however, are kept on the office list three months, and if desired must be renewed at the end of such period. Copy for the Bulletin must be in hand before the 12th of the month. The published list of "men available" is made up from members of the Society. Further information will be sent upon application.

POSITIONS AVAILABLE

801 Member, or member's son, with business ability and $10,000, can secure equal partnership with full financial control and management in new concern, to engage in the manufacture of a complete line of machinery fully covered by patents. Sales in the last three years, $100,000.

808 Young man of good education and foundry experience wanted for time-study and rate-setting work. Positions available in both steel and grey iron foundries. Location, St. Louis, Mo.

901 Partner with capital up to $20,000, to become associated with two expert mechanical engineers who have had broad experience in all English, French and American automobiles, and who desire to exploit a new engine.

904 Pennsylvania concern desires to communicate with competent men for position of assistant superintendent. Applicants must have thorough knowledge of both light and heavy machine production.

905 Young engineer of about 30 years of age wanted for planning, estimating and arranging for factory orders, of concern engaged in the manufacture of exterior and interior house furnishings, trim, and cabinet work.

906 Moderate sized plant manufacturing picture moldings desires superintendent; must be a manager of men and a good executive, thorough and energetic, with sufficient experience to show ability. In reply state salary expected, education, experience in full, age and references. Apply through the Society.

910 Assistant chief inspector. Technical graduate, with practical electrical and mechanical experience. If competent, would have charge of inspectors throughout the works. Applicant must be active, energetic, of highest moral character, and impartial, and a man of determination. Would be in direct line for promotion to chief inspector. About 30 years of age. Salary to start from $25 to $30 per week according to man's ability. Location, Massachusetts.

911 Assistant superintendent; American, 35 to 40 years of age; married; technical graduate, either mechanical or electrical, with practical experience in both branches; a good executive capable of handling manufacturing sections of works, employing about 1500 hands in manufacture of heating devices of all descriptions, enameled and cotton-covered

wire, lightning arresters and screw machine products. Must have experience on light sheet metal work, deep drawn work, small interchangeable manufacture, etc., to instruct tool department in kind of tools required, approximate cost of same, estimate probable cost of new products. Must be neat and orderly, energetic, and of the highest moral caliber. Salary $2000 to $3500 per year, according to applicant's past experience and record. Location, Massachusetts.

912 Foremen with experience on windings of various kinds, as motors, transformers, etc., with electrical experience, thoroughly posted on reading blue prints. Must be thorough executive, about 40 years of age, with about five years' experience in handling men and successful record in getting out work, both as to quality and quantity, without antagonizing workmen. Would probably be required to start as assistant foremen. Department would have from 60 to 125 men. Salary $27 to $32 per week. Location, Massachusetts.

914 Consulting engineer, Member of the Society and of the A.I.E.E., engaged in power and industrial work and general engineering practice in New York City, and extending through the southern states, desires partner who is not entirely dependent on present income and who can invest $1000. Applicant must be graduate mechanical or civil engineer with few years' experience in building construction and machinery installation, including steel and concrete work.

915 Mechanical engineer to take complete charge of drafting room, design of machinery, buildings and equipment. Location, New Jersey.

918 One or two technical men wanted on experimental and testing work in the steam engineering department of Buffalo concern. Work consists of general steel plant, inspecting, etc., such as furnaces, gas producers, engines and boilers, etc.

919 Man for permanent editorial position who knows power plant practice. Must be able to carry large responsibility and develop ideas. Young man preferred, willing to travel in western states. Apply through the Society.

921 Consulting engineer requires as assistant competent mechanical engineer having practical experience with mining machinery, coal-handling appliances and electrical installations. Must be competent to take charge of plants and carry out instructions as to operations, repairs and the purchase and installation of new equipment. Preferably a young man. Location, New York, but must be willing to travel. Moderate salary to begin, but exceptional opportunities if satisfactory. Apply through the Society.

923 Shop superintendent to take charge of the manufacturing operations for new Canadian plant making a general line of dairy machinery. Must have had experience in manufacturing. Apply through the Society.

MEN AVAILABLE

220 Mechanical engineer, Massachusetts Institute of Technology; three years' practical experience in power plant, efficiency engineering and concrete building construction.

221 Exclusive agency in New York for any kind of machinery desired by experienced engineer and salesman.

222 Member desires position as executive manager or general superintendent; practical and technical. Aggressive, broad gage, initiative, exceptional ability acquired by wide experience in manufacturing in plants employing 3000 men, which he has built and operated. Has tact and business ability, good organizer. Desires to connect with manufacturing, constructing or operating engineering company. Steam, hydraulic, electric and power installation work or in connection with consultation, examination, appraisal, design, organization, construction, operating, maintenance and commercial extension.

223 Member, graduate mechanical engineer, now employed, desires change; 15 years' experience in design, construction and maintenance of power and heating plants, industrial plant equipment, cost estimating, familiar with design and manufacture of automobiles, transmission and hoisting machinery; several years' shop training; competent to fill position of chief designer, factory engineer or superintendent; would consider teaching position with large engineering school.

224 Sales engineer desires position where knowledge of machinery and mill supply trade in United States and Canada is essential; seven years' varied experience, nine years in selling end. Experience in correspondence and design of selling contracts.

225 Member, 20 years' varied experience in designing, building and operating; now in charge of large construction work nearing completion; surface equipment of mines and gravel plant; has made special study of economical methods of handling materials; accustomed to handling men, organizing crews, drawing up contracts, designing and purchasing; desires connection with some firm handling large work. At present employed on Pacific coast.

226 Junior member, age 27, technical graduate, aggressive and energetic, employed by large steel company for past four years; familiar with operation and testing power plant machinery and maintenance of steel mill equipment; desires position with opportunity for advancement; at present employed. Middle West preferred.

227 Member desires position as mechanical superintendent in manufacturing plant; has held such position with large textile mills in New England for several years; conversant with mill construction, power plant installation and operation; electric transmission and general repair and care of large properties; first class references; holds first class steam engineer's license for Massachusetts.

228 Member having sold interest in eastern concern which he built up from a $10,000 business to $100,000 per year, is open for engagement. Experienced as machinist, marine engineer, tool, punch and die maker, designer, foreman, salesman and manager, including purchasing, selling and credits. 38 years old. Prefers middle, far west or foreign location.

229 Junior, mechanical engineer, technical graduate, Purdue University, 15 years' experience machine shop, drawing room, erecting large machinery, construction work and operating power plants of steam and gas engines.

230 Junior, 25 years of age, desires position with power company or contracting and consulting firm doing power plant work; technical graduate; experience in power plant of large manufacturing company.

231 Junior member, age 31, seven years' experience in boiler manufacture and general steel plate work, detailing, designing, testing, estimating and assistant manager; two years engine salesman; desires responsible position offering good opportunities; graduate of prominent engineering school; salary $2400.

232 Rensselaer graduate mechanical engineer, located since graduation with automobile concern, doing practical shop work; knowledge of Spanish; wishes better position.

233 Sales engineer with largest manufacturer of power and mining machinery in the Middle West desires to make a change. Excellent references can be furnished.

234 Member, long experience on light and medium weight manufacturing, desires position as works manager or general superintendent.

235 Junior, technical graduate, desires position with consulting engineer, manufacturing concern. Six years' experience in general machine shop work, power, heating and ventilating plant design and construction; expert on high and low-pressure piping; good knowledge of modern manufacturing methods; 32 years old, energetic and systematic; married. Best references; New York district preferred.

236 Graduate engineer, age 42. Member of Society and Associate of American Institute of Electrical Engineers; married, experience teaching electrical and mechanical engineering, construction of small and large buildings in wood, brick or concrete. Can take charge of any constructive work or head position in mechanical or electrical engineering in a college or industrial work, also charge of power plant work.

237 Member, graduate mechanical engineer, age 34, capable designer, practical foundry and shop man, desires position as engineer, superintendent, manager or assistant; broad engineering experience as mechanic and executive in plant, shop and office, in heavy machine building and light interchangeable manufacture; salary commensurate with position.

238 Technical graduate, age 40, with 20 years' experience in teaching, commercial testing, consulting and factory management; also two years selling; has been works manager for large concerns in United States and Canada; specialty factory building, equipment and organization.

239 Member, age 47, at present employed, desires permanent engagement as factory manager or superintendent; wide experience in varied lines of light manufacturing, such as typewriters, guns and special automatic machinery; resourceful and inventive.

240 Consulting engineer, electrical-mechanical; Member of the Society and of A.I.E.E., is open for engagement as teacher, instructor or lecturer in technical night schools in or near New York City.

241 Mechanical-electrical engineer, progressive diplomatic business man, with character, initiative and proven ability; a technical graduate and Member of the Society and A.I.E.E. Fifteen years' exceptional experience in the design, installation and operation of power and industrial plants, distribution systems (especially underground) and hydraulic work, including marine practice; drafting, original research, contracting, manufacturing, consulting engineering and the application of scientific management to the above operations; age 33, American, traveled throughout the

East, South, and Middle West. Now employed; desires responsible position or partnership with an established engineer in or near New York City. Salary is of secondary consideration.

242 Member, mechanical engineer, 37 years of age, 12 years' experience in harvester and implement business; served as draftsman, foreman, designer, master mechanic, assistant superintendent and for the past six years as general superintendent; familiar with modern and economical methods of manufacturing, establishing of efficiency by scientific rate-setting.

244 Member, age 37, at present manager of a general machine and structural shop, desires responsible position with machine works in Pennsylvania, eastern part preferred; is in position to make investment in responsible growing concern of this character.

245 Member, experienced as general or business manager, mechanical engineer, salesman and executive; desires position with manufacturing establishment or branch house where this experience will be useful; salary $3000.

246 Technical graduate, desires position requiring engineering, executive, or confidential ability, or to become associated with a firm of consulting, purchasing or inspecting engineers; practical manufacturing experience in iron and steel castings and machinery; can thoroughly satisfy anyone looking for a man of exceptional experience and ability.

247 Graduate mechanical engineer, age 36. At present employed in executive sales position, but desires to change for better and more permanent work in or near New York. Excellent experience in engineering, purchasing and sales work with consulting, manufacturing and selling concerns.

248 Mechanical engineer, age 29, technical graduate, desires position as assistant plant or mechanical engineer. Experienced in maintenance of large manufacturing plant, inspection, and design of electrical and mechanical equipment.

249 Superintendent or chief engineer for maintenance, design, construction and operation of industrial plants. East or Middle West preferred. Technical graduate.

251 Member, 35 years of age, graduate of Cornell University; wide experience in design, construction and operation of power plants and other machinery; design and superintendence of mill-building construction; desires position with large corporation as supervising engineer to take complete charge of all power operation, repairs and new construction work. At present employed and will consider only high-grade position.

252 Young engineer, five years' practical experience in design, erection and operation of power plants. Junior member.

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Note—Numbers in parentheses indicate number of years the member has yet to serve.

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Note—Numbers in parentheses indicate number of years the member has yet to serve.

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INSTITUTION	DATE AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	CHAIRMAN	CORRESPONDING SECRETARY
Armour Inst. of Tech.	Mar. 9, 1909	G. F. Gebhardt	H. E. Erickson	A. N. Koch
Case School of Applied Science	Feb 14, 1913	F. H. Vose	H. C. Mummert	C. Stemm
Columbia University	Nov. 9, 1909	Chas. E. Lucke	F. B. Schmidt	H. F. Allen
Cornell University	Dec. 4, 1908	R C. Carpenter	J. G. Miller	Edw. Mendenhall
Lehigh University	June 2, 1911	P B. de Schweinitz	W. C. Owen	T. G. Shaffer
Leland Stanford Jr. Univ.	Mar. 9, 1909	W. F. Durand	C. T. Keefer	K. J. Marshall
Mass. Inst. of Tech.	Nov. 9, 1909	E. F. Miller	W. H. Treat	L. L. Downing
New York University	Nov. 9, 1909	C. E Houghton		
Ohio State University	Jan.10, 1911	Wm. T. Magruder	R. H. Neilan	R. M. Powell
Penna. State College	Nov. 9, 1909	J. P. Jackson	H. L. Swift	H. L. Hughes
Poly. Inst. of Brooklyn	Mar. 9, 1909	W. D. Ennis	B. L. Huestis	A. Bielek
Purdue University	Mar. 9, 1909	G. A. Young	A. D. Meals	G. F. Lynde
Rensselaer Poly. Inst.	Dec. 9, 1910	A. M. Greene, Jr.	E. Kneass	R. F. Fox
State Univ. of Iowa	Apr.11, 1913	R. S. Wilbur	F. H. Guldner	C. S. Thompson
State Univ. of Kentucky	Jan.10, 1911	F. P. Anderson	R. R. Taliaferro	F. J. Forsyth
Stevens Inst. of Tech.	Dec. 4, 1908	Alex.C.Humphreys	L. F. Bayer	C. H. Colvin
Syracuse University	Dec. 3, 1911	W. E. Ninde	O. W. Sanderson	R. A. Sherwood
Univ. of Arkansas	Apr.12, 1910	B. N. Wilson	M. McGill	C. Bethel
Univ. of California	Feb.13, 1912	Joseph N. LeConte	J. F. Ball	G. H. Hagar
Univ. of Cincinnati	Nov. 9, 1909	J. T. Faig	A. O. Hurxthal	E. A. Oster
Univ. of Illinois	Nov. 9, 1909	W. F. M. Goss	A. H. Aagaard	H. E. Austin
Univ. of Kansas	Mar. 9, 1909	F. W. Sibley	E. A. Van Houten	L. E. Knerr
Univ. of Maine	Feb. 8, 1910	Arthur C. Jewett	E. H. Bigelow	O. H. Davis
Univ. of Missouri	Dec. 7, 1909	H. Wade Hibbard	W. P. Jesse	R. Runge
Univ. of Minnesota	May12, 1913	J. J. Flather		
Univ. of Nebraska	Dec. 7, 1909	J. D. Hoffman	A. A. Luebs	G. W. Nigh
Univ. of Wisconsin	Nov. 9, 1909	A. G. Christie	W. K. Fitch	J. W. Griswold
Washington University	Mar.10,1911	E. L. Ohle	D. Southerland	A. Schleiffarth
Yale University	Oct.11, 1910	L. P. Breckenridge	C. E. Booth	O. D. Covell

THE ADVENT OF A NEW YEAR

A new fiscal year of the Society begins October 1 and the Committee on Increase of Membership stands upon its threshold with just satisfaction over the results accomplished in the past year. As The Journal goes to press 1016 applications have been filed since October 1, 1912.

In no case has an invitation to join the Society been extended by the Committee to anyone not suggested by a member of the Society. Therefore the success of the work and the standing of the engineers invited has been dependent upon the response made by the membership. Not more than one-tenth of the members have thus far responded to the efforts of the Committee in this important work, and while much has been accomplished a great deal remains to be done, and it is earnestly hoped that the results already obtained will induce every member to shoulder his share of a work in which all can and should participate.

If each member will respond to this appeal during the ensuing year by obtaining the application of *at least one well-qualified engineer*, the added value of membership accruing to each individual will well repay the effort expended in this work.

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